PART 6

Air Handling Equipment

Carrier

CONTENTS PART 6 AIR HANDLING EQUIPMENT

SYSTEM DESIGN MANUAL

SUMMARY OF PART SIX

This part of the System Design Manual presents practical data and examples for selection and application of air handling equipment for normal air conditioning systems.

CHAPTER 1. FANS

This chapter presents information to guide the engineer in the practical application of fans used in air conditioning systems.

A fan is a device used to produce a flow of air. Use of the term is limited by definition to devices producing pressure differentials of less than 28 in. wg at sea level.

TYPES OF FANS

Fan are identified by two general groups:

1. Centrifugal, in which the air flows radially thru the impeller. Centrifugal fans are classified according to wheel blading ; forward – curved, backward–curved and radial (straight).
2. Axial flow, in which the air flows axially thru the impeller. Axial flow fans are classified as propeller (disc), tubeaxial and vaneaxial.

Figures 1, 2a, 2b and 2 c show the various types of commonly applied fans.

APPLICATION

When a duct system is needed in an air conditioning application, a tubeaxial, vaneaxial or centrifugal fan may be used. Where there is no duct system and little resistance to air flow, a propeller fan can be applied. However, self – contained equipment often utilizes centrifugal fans for applications without ductwork.

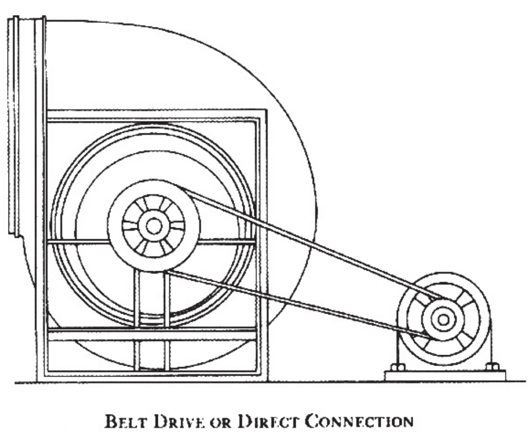


Fig. 1 – Centrifugal Fan

The centrifugal fan is used in most comfort applications because of its wide range of quiet, efficient operation at comparatively high pressures. In addition, the centrifugal fan inlet can be readily attached to an apparatus of large cross – section while the discharge is easily connected to relatively small ducts. Air flow can be varied to match air distribution system requirements by simple adjustments to the fan drive or control devices.

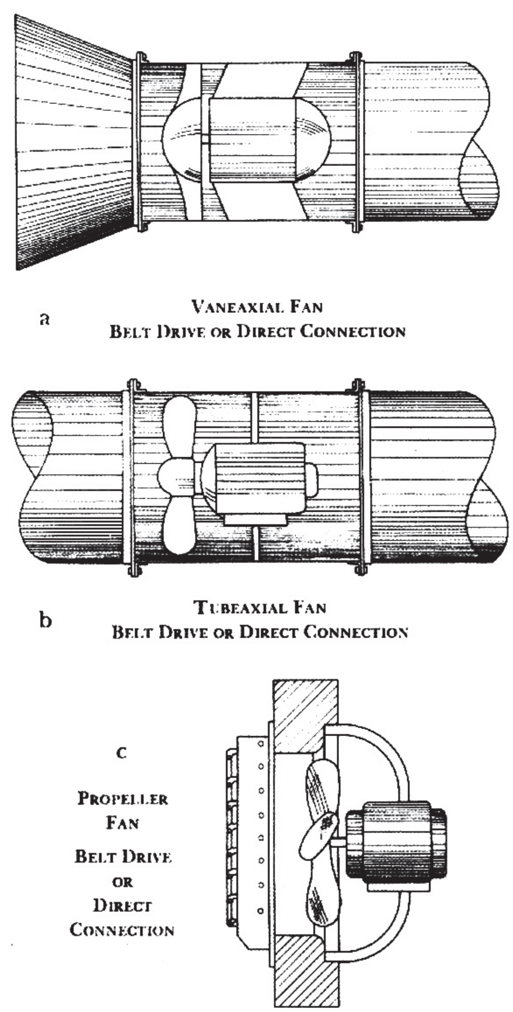


Fig. 2 – Axial Flow Fans

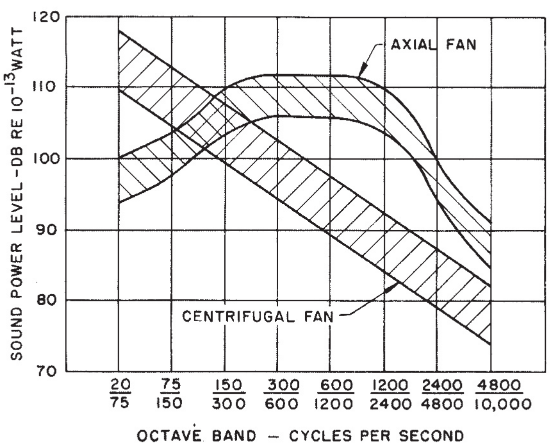
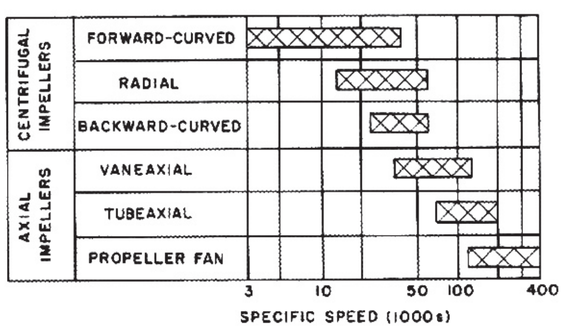


Fig. 3 – Sound Power Levels

Axial flow fans are excellent for large air volume applications where higher noise levels are of secondary. They are, therefore, often used for industrial air conditioning and ventilation. These high velocity fans require guide vanes to obtain the best efficiencies when operation against pressures considered normal for centrifugal fan . However, these fans may be applied without guide vanes.

Figure 3 illustrates the approximate sound power level of a typical centrifugal fan and an axial flow fan. The frequencies detectable by the human ear (300 to 10,000 cycles per second) are the least favorable for the axial flow fan. Therefore, to obtain acceptable sound levels with the axial flow fan, sound attenuation may be required.



Courtesy of The Torrington Manufacturing Co.

NOTE: $$\text{Specific speed} = \frac{\text{rpm} \times (\text{cfm})^{\frac{1}{2}}}{(\text{static pressure})^{\frac{3}{4}}}$$

Fig. 4 – Specific Speed Ranges

The concept of specific speed is useful in describing the applications of various fan types. Specific speed is a fan performance index based on the fan speed, capacity and static pressure. *Figure 4* shows the ranges of specific speed in which six types of centrifugal and axial flow fans operate at high static efficiencies. This figure indicates that forward – curved blade centrifugal fans attain their peak efficiencies at low speeds, low capacities and at high static pressures. However, propeller fans reach high efficiency at high speeds and capacities and at low static pressures.

The horsepower characteristics of the various fans are such that a type may be overloading or non overloading. The backward – curved blade centrifugal fan is a nonoverloading type. The forward – curved and radial blade centrifugal fans may overload. Axial flow fans may be either nonoverloading or overloading.

All fan types may be utilized for exhaust service. Wall fans operate against little or no resistance and therefore are usually of the propeller type. Propeller fans are sometimes incorporated into factory – built penthouses or roof caps. Hooded exhaust fans and central station exhaust fans are typically of the centrifugal type. Axial fans may be suitable for exhaust applications, particularly in factory installations.

STANDARDS AND CODES

Fan application and installation should conform to all codes, laws and regulations applying at the job site.

The AMCA standard Test Code for Air Moving Devices, Bulletin 210, prescribes methods of testing fans, while AMCA rating standards prescribe methods of rating.

CENTRIFUGAL FANS

Centrifugal fans are identified by the curvature of the blade tip. The forward – curved blade curves in the direction of rotation (*Fig. 5a*) . The radial blade has no curvature (*Fig. 5b*). The backward–curved blade tip inclines backward, curving away from the direction of rotation (*Fig 5c*) . The curvature of the blade tip defines the shape of the horsepower and static pressure curves.

The characteristics of the three main types of centrifugal fans are listed in Table 1.

FORWARD – CURVED BLADE FAN

A typical performance of a forward – curved blade fan is shown in *Fig. 6* . The pressure rises from 100 % free delivery toward no delivery with a characteristic dip at low capacities. Horsepower increases continuously with increasing air quantity.

BACKWARD – CURVED BLADE FAN

A typical performance of a backward – curved blade fan is shown in *Fig. 7*. The pressure rises constantly from 100 % free delivery to nearly no delivery. There is no dip in the curve. The horsepower curve peaks at high capacities. Therefore, a motor selected to satisfy the maximum power demand at a given fan speed does not overload at any point on the curve, providing this speed is maintained.

Two modifications of the backward – curved blade fan are the airfoil and backward – inclined blade fans.

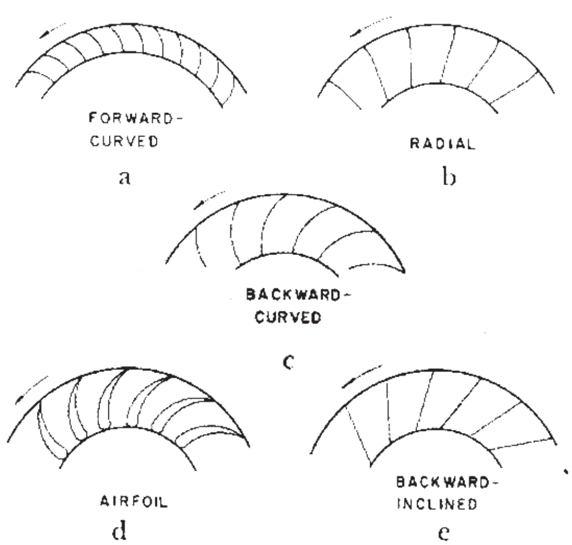


Fig. 5 – Fan Blades

TABLE 1—CHARACTERISTICS OF CENTRIFUGAL FANS

FAN TYPE	ADVANTAGES
Forward-Curved	1. Runs at a relatively low speed compared to other types for the same capacity. 2. Smaller fan for a given duty, excellent for fan-coil units.
Radial	1. Self-cleaning. 2. Can be designed for high structural strength to achieve high speeds and pressures.
Backward-Curved	1. More efficient. 2. Horsepower curve has a flat peak so that the motor may be sized to cover the complete range of operation from zero to 100% air flow for a single speed. Nonoverloading. 3. Pressure curve is generally steeper than that of the forward-curved fan. This results in a smaller change in air volume for any variation in system pressure for selections at comparable percentages of free delivery. 4. Point of maximum efficiency is to the right of the pressure peak, allowing efficient fan selection with a built-in pressure reserve. 5. Quieter than other types.

These are illustrated in *Fig. 5 d and 5 e* . Both are nonoverloading types.

The airfoil blade fan is a high efficiency fan because its aerodynamically shaped blades permit smoother airflow thru the wheel. It is normally used for high capacity, high pressure applications where power savings may outweigh its higher first cost. Since the efficiency characteristic of an airfoil blade fan usually peaks more sharply than those of other types, greater care is required in its selection and application to a particular duty.

The backward – inclined blade fan must be selected closer to free delivery; therefore, it does not have as

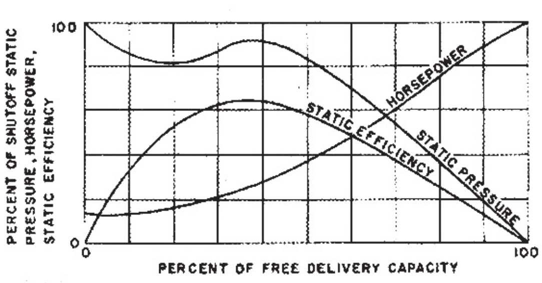


Fig. 6 – Forward-Curved Blade Fan Performance

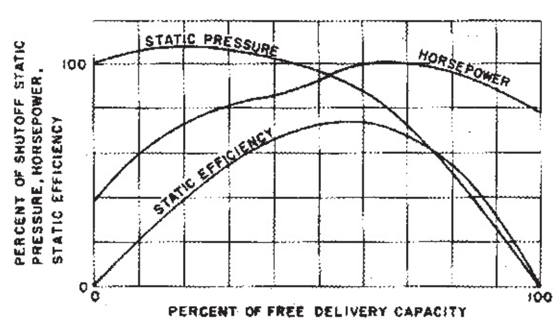


Fig. 7 - Backward-Curved Blade Fan Performance

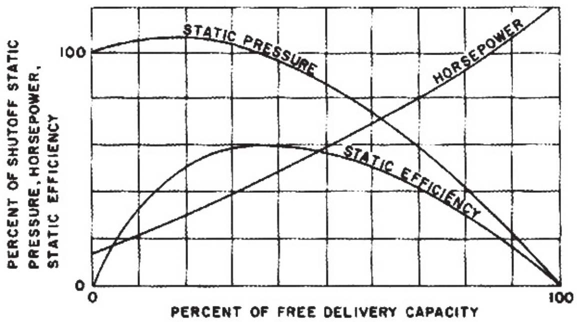


Fig. 8 – Radial Blade Fan Performance

great a range of high efficiency operation as does the backward –curved blade fan. Manufacture of an inclined blade is understandably a simpler operation.

RADIAL BLADE FAN

Typical performance of a radial (straight) blade fan is shown in *Fig. 8*. The pressure characteristic is continuous at all capacities. Horsepower rises with increasing air quantity in an almost directly proportional relation. Thus, with this type of fan the motor may be overloaded as free air delivery is delivery is approached.

The radial blade fan has efficiency, speed and capacity characteristics that are midway between the forward–curved and backward–curved blade fans. It is seldom used in air conditioning applications because it lacks an optimum characteristic.

AXIAL FLOW FANS

Figure 9 shows a performance characteristic typical of a propeller fan.

The tubeaxial fan is a common axial flow fan in a

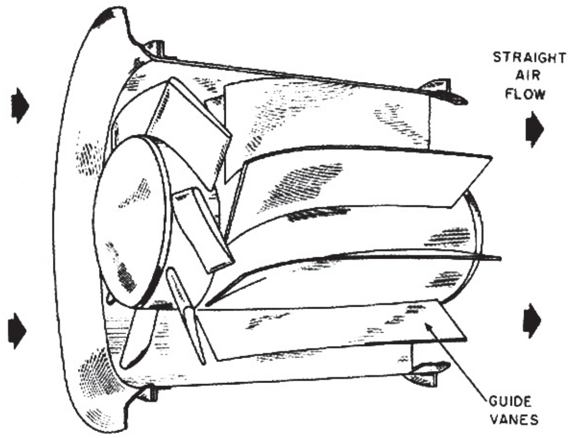


Fig. 10 – Vaneaxial Fan

tubular housing but without inlet guide vanes. The blade shape may be flat or curved, of single or double thickness.

The axial flow fan has become particularly associated with the vaneaxial type which has guide vanes before or after the fan wheel. To make more effective use of the guide vanes, the fan wheel usually has curved blades of single or double thickness. *Figure 10* is a sectional view of the vaneaxial fan.

The curved stationary diffuser vanes are the type most frequently used when higher efficiency vaneaxial fans are desired. The purpose of these vanes is to recover a portion of the energy of the tangentially accelerated air.

Typical performance of an axial flow fan is shown in *Fig.11.*

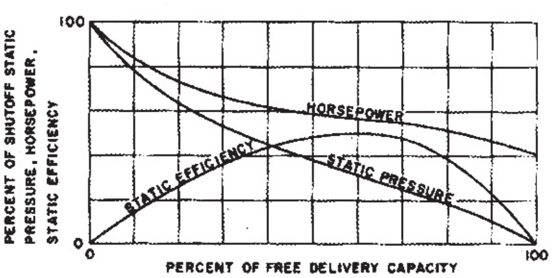


Fig. 9 – Propeller Fan Performance

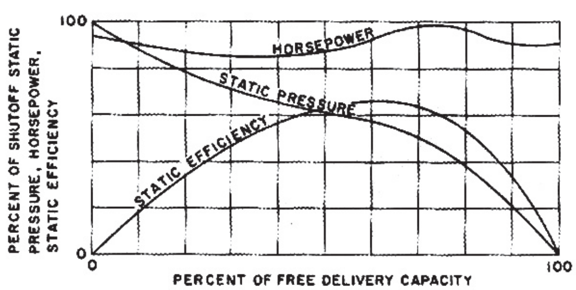


Fig. 11 – Axial Flow Fan Performance

FAN DESIGNATION

CLASS OF CONSTRUCTION

The AMCA has developed standards of centrifugal fan construction based on the pressure the fans are required to develop. The four classes of fans appear in Table 2 , Each of the various fan manufacturers has defined his own maximum wheel tip speed for each class.

The required fan class can be determined from *Chart 1* it outlet velocity and static pressure are know. Calculation of velocity pressure and total pressure is thus eliminated. This chart is based on standard air (29.92 in. Hg barometric pressure and 70 F temperature).

If nonstandard conditions of temperature and altitude are encountered in an application, the calculated static pressure should be corrected before entering Chart 1. This procedure is described in the section entitled *Fan Selection. See Example 3.*

Minimum first costs can often be achieved by using a larger size fan of a given class than by choosing a smaller fan size of a higher class. If a selection lies on the border line, both alternatives should be considered.

TABLE 2—CLASSES OF CONSTRUCTION

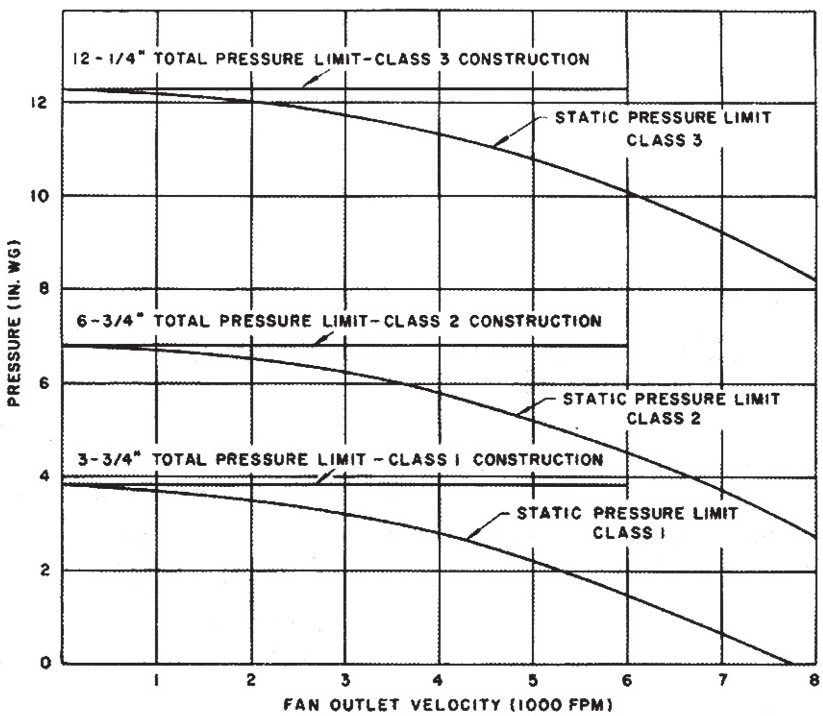
Centrifugal Fans

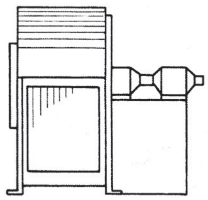
CLASS	MAXIMUM TOTAL PRESSURE
I	3¾ in. wg — standard
II	6¾ in. wg — standard
III	12¾ in. wg — standard
IV	More than 12¾ in. wg — recommended

Some manufacturers offer packaged fans and motors which are not defined in terms of classes. These packages are made of Class I or II parts, modified slightly to hold the motor within the fan base. The fan package is less expensive than the equivalent Class I or II fan and is satisfactory for most applications. Packaged fans are also offered in construction lighter than Class I. Manufacturers' specifications usually distinguish between light and heavy construction.

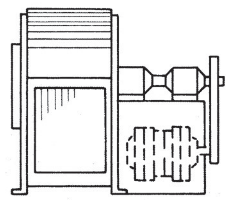
A pressure class standard pertaining to centrifugal fans mounted in cabinets has also been published by AMCA. Cabinet fans are commonly used with central station fan – coil equipment. The three classes of such fans are defined in *Table 3* .

CHART 1 – CONSTRUCTION CLASS PRESSURE LIMITS

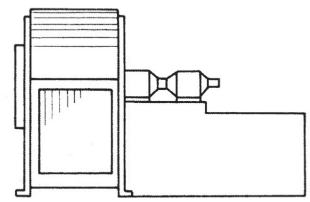




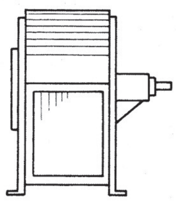
Arrangement 1, SW, SI
For belt drive or direct connection. Wheel overhung. Two bearings on base.



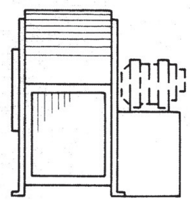
Arrangement 9, SW, SI
For belt drive. Arrangement 1 designed for mounting prime mover at side of base.



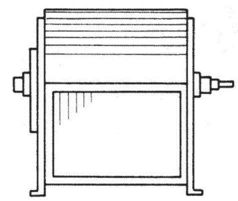
Arrangement 8, SW, SI
For belt drive or direct connection. Arrangement 1 plus base for prime mover.



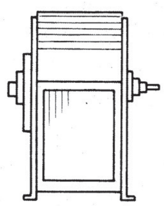
Arrangement 2, SW, SI
For belt drive or direct connection. Wheel overhung. Bearings in bracket supported by fan housing.



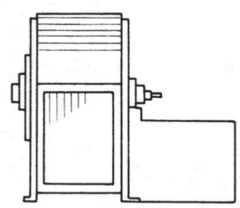
Arrangement 4, SW, SI
For direct drive. Wheel overhung on prime mover shaft. No bearings on fan. Base mounted or an integrally direct connected prime mover.



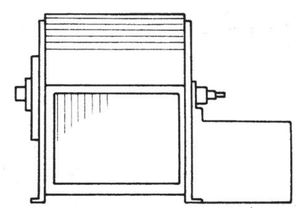
Arrangement 3, DW, DI
For belt drive or direct connection. One bearing on each side and supported by fan housing.



Arrangement 3, SW, SI
For belt drive or direct connection. One bearing on each side and supported by fan housing. Not recommended on 27 in. or smaller diameter wheel.



Arrangement 7, SW, SI
For belt drive or direct connection. Arrangement 3 plus base for prime mover. Not recommended on 27 in. or smaller diameter wheel.



Arrangement 7, DW, DI
For belt drive or direct connection. Arrangement 3 plus base for prime mover.

FIG. 12 — DRIVE ARRANGEMENTS

TABLE 3—CLASSES OF CONSTRUCTION
Cabinet Fans

CLASS	MAXIMUM STATIC PRESSURE
A	3 in. wg
B	5½ in. wg
C	More than 5½ in. wg

Fan class nomenclature does not apply to fans used in fan–coil terminal units where the manufacturer limits such fans to a particular maximum speed and static pressure.

FAN ARRANGEMENTS

Centrifugal fan drive arrangement , standardized by AMCA , refers to the relation of the fan wheel to the bearings and the number of fan inlets. *Figure 12* indicates the various arrangements.

The fan drive may be direct of by belt. With the exception of packaged fans and motors, direct drive is seldom employed in air conditioning applications because of the greater flexibility afforded by the belt drive.

Arrangements 1 , 2 and 3 are commonly used for air conditioning. The remaining choices are modified versions of Arrangements 1 and 3 . Double inlet fans for belt drive are available in Arrangement 3 and 7.

In selecting a suitable fan arrangement first cost and space requirements are considered. Single inlet while double inlet fans are lower in cost in the large sizes. For the same capacity a single inlet fan is about 30 % taller than the double inlet type, but only about 70 % as wide.

Arrangement 3 is the most widely used because the bearing location eliminates the necessity for a bearing platform. Cost and required space is therefore minimized.

For single inlet applications Arrangements 1 and 2 are used where the fan wheel is less than 27 inches in diameter. Arrangement 3 is not used since the bearing on the inlet side is large enough, relative to the inlet area, to affect fan performance. Fans of larger sizes and double inlet fans are not limited in this way.

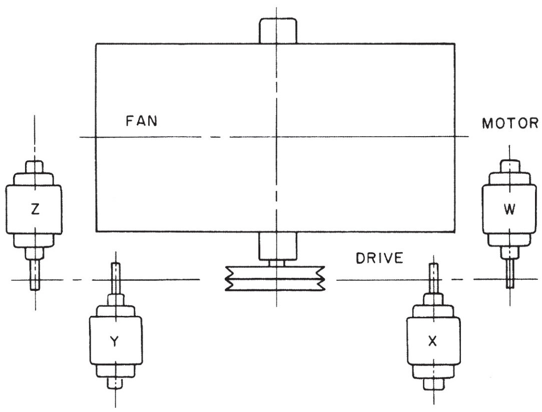
Arrangement 1 is usually more costly than Arrangement 2 because it has two bearings and a base. Where Class III construction is required, Arrangement 1 is preferred over Arrangement 2.

If Arrangement 3 is to be used at air temperatures exceeding 200 F or Arrangement 1 or 2 at temperatures exceeding 300 F, the fan manufacturer should be consulted so that the proper bearing or heat slinger can be specified.

Table 4 compares the costs of fan and drive for several single inlet arrangements. Selections are based on a constant air quantity and static pressure.

TABLE 4—ARRANGEMENT COST COMPARISON

ARRANGEMENTS	MATERIAL COSTS (%)
1	117
2	100
3	100
9	124

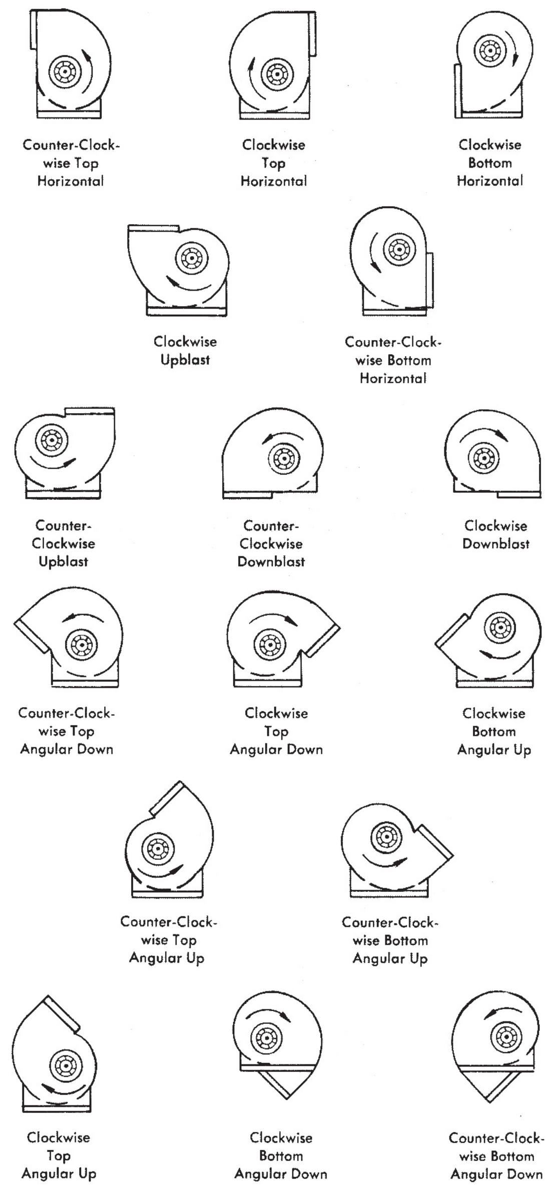


The location of the motor is determined by facing the drive side of the fan or blower and designating the motor position by the letters W, X, Y or Z as necessary. (NOTE: This designation is used when ordering isolation bases.)

Fig. 13 – Motor Positions

Figure 13 shows the motor positions possible for a belt – driven fan. Use of Positions W and Z results in the simplest construction of fan base and belt guard.

Figure 14 shows the standard rotation and discharge combinations available.



The direction of rotation is determined from the driving side for both single and double inlet fans. (The driving side of a single inlet fan is considered to be the side opposite the inlet, regardless of the actual location of the drive.) When fans are to be inverted for ceiling suspension, the direction is determined when the fan is resting on the floor.

Fig. 14 – Rotation and Discharge

Axial flow fans are available for belt drive or direct connections. Therefore, two arrangements have been standardized throughout the industry, Arrangement 4 is driven directly, Since the motor is in the air stream, the application of this arrangement is limited to the handling of air which will not damage the motor. Arrangement 9 is belt – driven, with the motor located outside of the air stream and the drive protected.

FAN PERFORMANCE

Fan performance curves show the relation of pressure, power input and fan efficiency for a desired range of air volumes. This relation is based on constant speed and air density.

Static rather than total pressure and efficiency are usually inferred. Static pressure best represents the pressure useful in overcoming resistance. However, static pressure is less applicable where the fan outlet velocity is high. Further, if the fan operates against no resistance, static pressure is meaningless. In these cases total mechanical efficiency is used.

Fan performance may be expressed as percentages of rated quantities or in terms of absolute quantities. The former method is illustrated in *Fig. 6 , 7 , 8 , 9 and 11*.

LAWS OF FAN PERFORMANCE

Fan laws are used to predict fan performance under changing operating condition or fan size.

They are applicable to all types of fans.

The fan laws are stated in *Table 5*. The symbols used in the formulas represent the following quantities:

- *Q* – Volume late of flow thru the fan.
- *N* – Rotational speed of the impeller.
- *P* – Pressure developed by the fan, either static or total.
- *Hp* – Horsepower input to the fan.
- *D* – Fan wheel diameter. The fan size number may be used if it is proportional to the wheel diameter.
- *W* – Air density, varying directly as the barometric pressure and inversely as the absolute temperature.

In addition to the restrictions noted in *Table 5*, application of these laws is limited to cases where fans are geometrically similar

TABLE 5—FAN LAWS

VARIABLE	CONSTANT	NO.	LAW	FORMULA
SPEED	Air Density Fan Size Distribution System	1	Capacity varies as the Speed.	$\frac{Q_1}{Q_2} = \frac{N_1}{N_2}$
		2	Pressure varies as the square of the Speed.	$\frac{P_1}{P_2} = \left(\frac{N_1}{N_2}\right)^2$
		3	Horsepower varies as the cube of the Speed.	$\frac{Hp_1}{Hp_2} = \left(\frac{N_1}{N_2}\right)^3$
FAN SIZE	Air Density Tip Speed	4	Capacity and Horsepower vary as the square of the Fan Size.	$\frac{Q_1}{Q_2} = \frac{Hp_1}{Hp_2} = \left(\frac{D_1}{D_2}\right)^2$
		5	Speed varies inversely as the Fan Size.	$\frac{N_1}{N_2} = \frac{D_2}{D_1}$
		6	Pressure remains constant.	$P_1 = P_2$
	Air Density Speed	7	Capacity varies as the cube of the Size.	$\frac{Q_1}{Q_2} = \left(\frac{D_1}{D_2}\right)^3$
		8	Pressure varies as the square of the Size.	$\frac{P_1}{P_2} = \left(\frac{D_1}{D_2}\right)^2$
		9	Horsepower varies as the fifth power of the Size.	$\frac{Hp_1}{Hp_2} = \left(\frac{D_1}{D_2}\right)^5$
AIR DENSITY	Pressure Fan Size Distribution System	10	Speed, Capacity and Horsepower vary inversely as the square root of Density.	$\frac{N_1}{N_2} = \frac{Q_1}{Q_2} = \frac{Hp_1}{Hp_2} = \left(\frac{W_2}{W_1}\right)^{1/2}$
	Capacity Fan Size Distribution System	11	Pressure and Horsepower vary as the Density.	$\frac{P_1}{P_2} = \frac{Hp_1}{Hp_2} = \frac{W_1}{W_2}$
		12	Speed remains constant.	$N_1 = N_2$

and where there is no change in the point of rating on the performance curves. Because of the latter qualification, fan efficiencies are assumed constant.

Geometrically similar fans are those in which all dimensions are proportional to fan wheel diameter.

The same point of rating for two fans of different size means that for each fan the pressure and air volume at the point of rating are the same fraction of shutoff pressure and volume at free delivery, provided the rotational speed is the same in either case. For example, an operating point on *Fig.7* will not change with the application of laws 7 thru 9 , even though specific values will change.

Example 1 – use of Laws 1 thru 3

Given:

Air quantity	- 33,120 cfm
Static pressure	- 1.5 in wg
Fan speed	- 382 rpm
Brake horsepower	- 10.5

Find:

Capacity, static pressure and horsepower if the speed is incased to 440 rpm.

Solution:

Capacity = 33,120 x (440 / 382) = 38,150 cfm

Static pressure = $1.5 \times (440 / 382)^2 = 2.0$ in. wg

Horsepower = $10.5 \times (440 / 382)^3 = 16.1$ bhp

FAN CURVE CONSTRUCTION

Fan performance is usually presented in tabular form (*Table 6*). However, for a graphic analysis performance curves are more convenient to use. If no curves are available, tabular values of pressure and horsepower may be plotted at constant speeds over the given range of capacities, The resulting curves may then be used as described under *Fan Performance in a System.*

FAN PERFORMANCE IN A SYSTEM

SYSTEM BALANCE

Any air handling system consists of a particular combination of ductwork, heaters, filters, dehumidifiers and other components. Each system therefore has an individual pressure–volume characteristic which is independent of the fan applied to the system. This relation may be expressed graphically on a coordinate system identical to that of a fan performance curve. A typical system characteristic is shown in *Fig. 5*.

System curves are based on the law which states that the resistance to air flow (static pressure) of a system varies as the square of the air volume flowing thru the system. In practice a static pressure is calculated as carefully as possible for a given system at the required air quantity. This establishes one point of the system curve. The remaining points are obtained by calculation from the above law, rather than by further static pressure calculations at other air quantities.

When a fan performance curve for a given fan size and speed is superimposed upon a system characteristic as in *Fig. 15* , there is the only possible operating point under the conditions. If the fan speed is in decreased, the point of operation moves upward toward the right. If the speed is decreased, the operating point moves down and the to the left.

Figure 15 illustrates the effect on system performance of operation at other than design conditions. Such a situation could be caused by dirty filters, wet coil versus dry coil operation of the dehumidifier, or the modulation of a damper. Lines of constant brake horsepower have been included for ease of analysis.

TABLE 6—TYPICAL FAN TABLE

WHEEL DIAM. 44½″ — INLET AREA = 21.60 — **BACKWARD-CURVED**

TIP SPEED = 11.65 × RPM — OUTLET AREA = 20.79 — Class I Ratings

CFM	Outlet Veloc.	¼″ SP		⅜″ SP		½″ SP		⅝″ SP		¾″ SP		⅞″ SP		1″ SP		1¼″ SP		1½″ SP		1¾″ SP	
	FPM	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP
12460	600	189	.61																		
14536	700	200	.76	228	1.04																
16613	800	212	.94	237	1.26	264	1.64														
18691	900	228	1.13	250	1.49	272	1.87	292	2.25												
20766	1000	243	1.39	263	1.76	282	2.18	302	2.57	324	3.08										
22841	1100	259	1.66	278	2.09	296	2.52	315	2.95	332	3.44	351	3.98								
24916	1200	276	1.98	294	2.45	310	2.92	327	3.38	343	3.87	359	4.37	377	4.91						
26991	1300	292	2.34	309	2.84	325	3.35	342	3.85	356	4.36	371	4.88	387	5.44	420	6.66				
29065	1400	309	2.77	325	3.29	342	3.83	356	4.37	370	4.91	384	5.47	398	6.03	426	7.22	459	8.62		
31158	1500	326	3.24	342	3.82	357	4.37	371	4.95	385	5.53	398	6.12	411	6.71	437	7.92	465	9.25	492	10.53
33233	1600	344	3.78	359	4.37	373	4.97	387	5.58	400	6.19	413	6.80	425	7.45	449	8.73	473	10.01	500	11.45
35308	1700	361	4.37	375	5.00	390	5.63	403	6.28	416	6.93	427	7.58	440	8.24	463	9.59	486	10.93	509	12.42
37383	1800	379	5.04	394	5.69	407	6.37	420	7.04	431	7.72	443	8.41	454	9.11	477	10.53	498	11.95	520	13.45
39458	1900	397	5.76	411	6.46	423	7.16	436	7.88	447	8.60	459	9.31	470	10.04	492	11.54	512	13.03	533	14.58
41532	2000	416	6.57	428	7.31	441	8.05	452	8.78	464	9.54	475	10.30	486	11.05	507	12.60	526	14.15	546	15.75

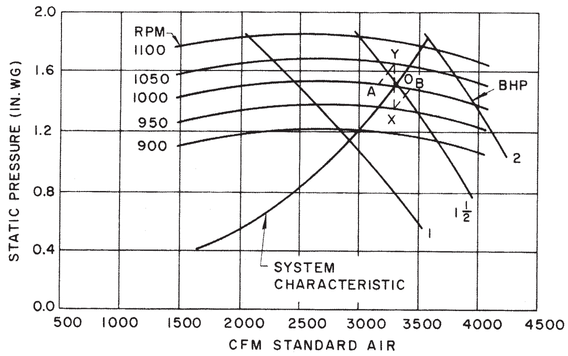


Fig. 15 – Effect of Change in Design Conditions

Example 2 describes the analysis.

Point *O* is the design point. Points *A* and *B* are new operating points resulting respectively from an increase or decrease in system resistance. Point *A* and *B* are single points each of two new system characteristics.

Example 2 – Operation Above Design Static Pressure

Given:

Air quantity	- 3300 cfm
Static pressure	- 1.5 in. wg
Fan speed	- 1000 rpm
Fan performance	- *Fig . 15*

Find:

Air quantity, static pressure and brake horsepower if the resistance of the filters is 0.15 in. wg greater than estimated for design.

Solution:

1. From design point O (*Fig. 15*) rise vertically to point *Y* at a static pressure of 1.65 in . wg (1.5 + 0.15) and 3300 cfm.
2. Since the fan operates at 1000 rmp, proceed to the 1000 rpm fan curve along a line parallel to the system characteristic. At the new operating point A the fan delivers to the system, 3175 cfm at a static pressure of 1.5 in. wg. The required power input at the new conditions is 1.4 bhp.

PRESSURE CONSIDERATIONS

An analysis similar to that of *Example 2* indicates that overestimating the static pressure increases the required horsepower. Operation in this case is at point B of *Fig. 15* , rather than at point *O* . Therefore, the addition of a safety factor to the calculated static pressure tends to increase fan horsepower requirements unnecessarily. The static pressure used to select a fan should be that pressure calculated for the system at the design air quantity.

If the static pressure is overestimated, the amount of increase in horsepower and air volume depends upon the

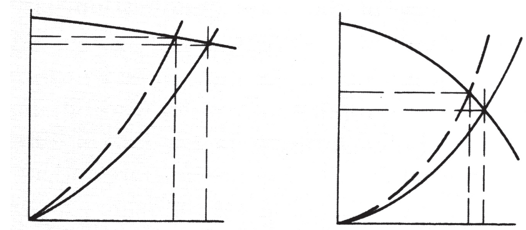


Fig. 16 — Effect of Fan Curve Slope

steepness of the fan curves in the area of selection. *Fig. 16* shows that volume deviations may be large if the fan curve is relatively flat. With a steep pressure characteristic, pressure differences may have little effect on air volume and horsepower. For this reason a fan with a steep performance curve is well suited to a system requiring an air volume relatively independent of changes in system resistance. An example of such a system is an induction unit primary air system.

Conversely, a variable volume system requires a pressure nearly constant with changes in air volume. Thus, a fan with a comparatively flat pressure characteristic is more appropriate.

STABILITY

Fan operation is stable if it remains unchanged after a slight temporary disturbance or if a slight permanent disturbance produces only a small shift in the operating point.

Instability is a surging or pulsation which may occur when the system characteristic curve intersects the fan curve at two or more points. This is rare occurrence in a single fan. When two or more forward curved fans are connected in parallel, It is possible that the composite curve have an unstable area such as shown in Fig. 17 . If the operating point falls in this area, either adding or subtracting resistance allows operation at a stable point on either side of this area. When operation occurs such that only one sharp intersection of fan curve and system curve is possible, there is no condition of instability.

System resonance is rare thing but may occur in systems utilizing high pressure fans with a duct system turned to a particular frequency like an organ pipe. With operation to the left of the pressure peak, a pressure increase is accompanied by a capacity increase, in turn tending to further increase pressure. This condition may be altering the system characteristic curve so that the operating point falls between the pressure peak and the free delivery point.

FAN SELECTION

The system requirements which influence the selection of a fan are air quantity, static pressure, air density if other than standard , prevailing sound level or the use of the space served, available space, and the nature of the load. When these requirements are known, the selection of a fan for air conditioning usually involves choosing the most inexpensive combination of size and class of construction with an acceptable sound level and efficiency.

Outlet velocity cannot be used as a criterion of selection from the standpoint of sound generation. The best sound characteristics are obtained at maximum fan efficiency. Fans operating at higher static pressures have greater allowable outlet velocities since maximum efficiency occurs at higher air quantities. Thus, any limits imposed on outlet velocity in relation to sound level depend upon the static pressure in addition to ambient sound levels and the use of the area served. In regard to sound generation a fan should be selected as near to maximum efficiency as is possible and adjacent ductwork should be properly designed, as described in *Part 2* .

The best balance of first cost and fan efficiency usually results with a fan selection slightly smaller than that representing the maximum efficiency available. However, selection of a larger, more efficient fan may be justified in the case of long operating hours. Also, a larger fan may be economically preferable if a smaller selection necessitates a larger motor, drive and starter, or heavier construction.

The selection of a fan and drive can affect psychrometric conditions in the area served. If the combination produces an air quantity below that required at design conditions, the resulting room dry - bulb temperature is higher. When the air quantity is greater than required at design conditions, room controls prevent a fall in temperature.

ATMOSPHERIC CORRECTIONS

Fan sound level does not vary sufficiently with altitude to warrant using sound ratings at conditions other than sea level.

Fan tables and curves are based on air at standard atmospheric conditions of 70 F and 29.92 in. Hg barometric pressure. If fan is to operate at nonstandard conditions, the selection procedure must include a correction. With a given capacity and static pressure at operating conditions the adjustments are made as follows:

1. Obtain the air density ratio from *Chart. 2*
2. Calculate the equivalent static pressure by dividing the given static pressure by the air density ratio.
3. Enter the fan tables at the given capacity and equivalent static pressure to obtain speed and brake horsepower. This speed is correct as determined.
4. Multiply the tabular brake horsepower by the air density ratio to find the brake horsepower at the operating conditions.

If atmospheric corrections are ignored in the fan selection, fan speed and air capacity may be too small, and the brake horsepower somewhat high.

Example 3 illustrates a fan selection at high altitude.

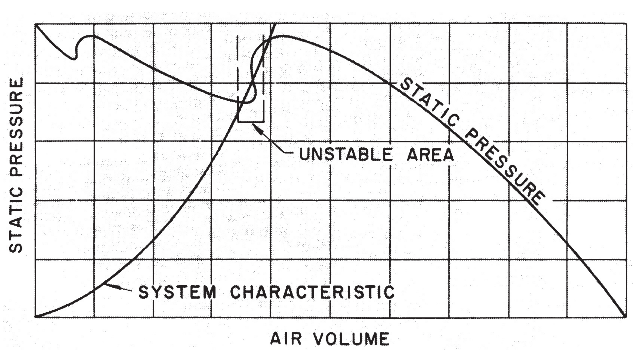
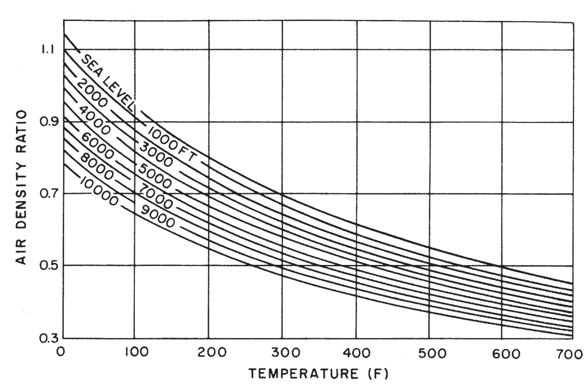


Fig. 17 – System Instability

CHART 2 – ATMOSPHERIC CORRECTIONS



NOTE: Air density ratio $= \dfrac{\text{density at new condition}}{\text{density of standard air}}$

Example 3 – Fan Selection at High Altitude

Given:

Air quantity*	- 37,380 cfm
Static pressure*	-1.45 in. wg
Altitude	- 5000 ft
Air temperature	- 70 F
Fan ratings	- *Table 6*

Find:

Fan speed, brake horsepower and class.

Solution:

1. From Chart 2 the air density ratio is 0.83.
2. The equivalent static pressure is equal to 1.45 / 0.83 or 1.75 in. wg.
3. Select from *Table 6* a size 7 double inlet fan, operating at 520 rpm and requiring 13.45 bhp.
4. The design speed at 5000 ft is 520 rpm.
5. The brake horsepower for the less dense air at 5000 ft is 0.83 x 13.45 or 11.2 bhp.
6. At the fan outlet velocity of 1800 fpm and the equivalent static pressure of 1.75 in. wg, enter Chart 1. The selection is well within the range of a Class I fan. This is the proper selection.

At altitudes above 3300 feet, fan motor temperature guarantees may not apply. High altitude applications should therefore be brought to the manufacturer's attention.

ACCESSORIES

Fan accessories are available to fulfill specific needs. Where applicable, the following accessories can aid in assuring a satisfactory air conditioning system.

Access Doors

Access doors on the fan scroll sheet should be provided whenever there is a possibility of dirt collecting the fan.

Drains

A drain should be specified whenever condensation or water carry – over may occur.

Variable Inlet Vanes

Figure 18 shown a set of variable inlet vanes. These vanes are a volume control device and may be automatically or manually actuated. They are recommended for applications with long periods of reduced capacity operation and for use with static pressure regulators. Use of variable inlet vanes is further discussed under Control.

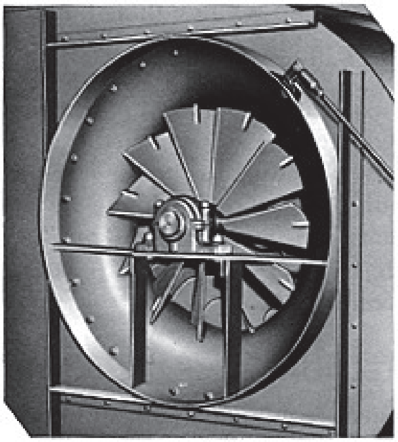


Fig. 18 – Variable Inlet Vanes

Outlet Dampers

Outlet dampers are a volume control device and may be automatically or manually actuated. They may be used for applications requiring extreme capacity reduction for short periods of time or for small adjustments. These dampers are illustrated in *Fig. 19*. Their use is further discussed under *Control*.

Isolators

In order of decreasing vibration isolation efficiency, steel spring isolators, double rubber – in shear isolators, and single rubber – in shear isolators are all used for fan installations. These isolators are normally used in conjunction with steel channel bases so that the fan and the motor may be mounted on an integral surface. For a more complete discussion of vibration isolation, refer to *Chapter 2* of this part.

Bearings

Ball bearings are the most common type of bearing used on fans. The sleeve oil bearing can be provided at an extra cost and is initially a quieter bearing. However, its quietness has been overemphasized since the noise does not materially add to the fan air noise.

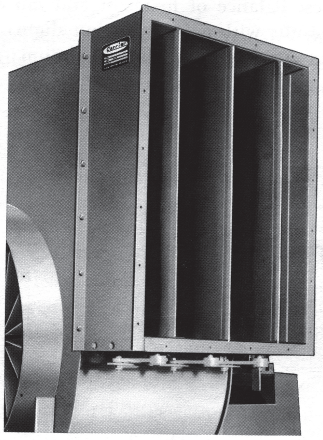


Fig. 19 – Outlet Dampers

CONTROL

Variation of the air volume delivered by a fan may be accomplished by several methods:

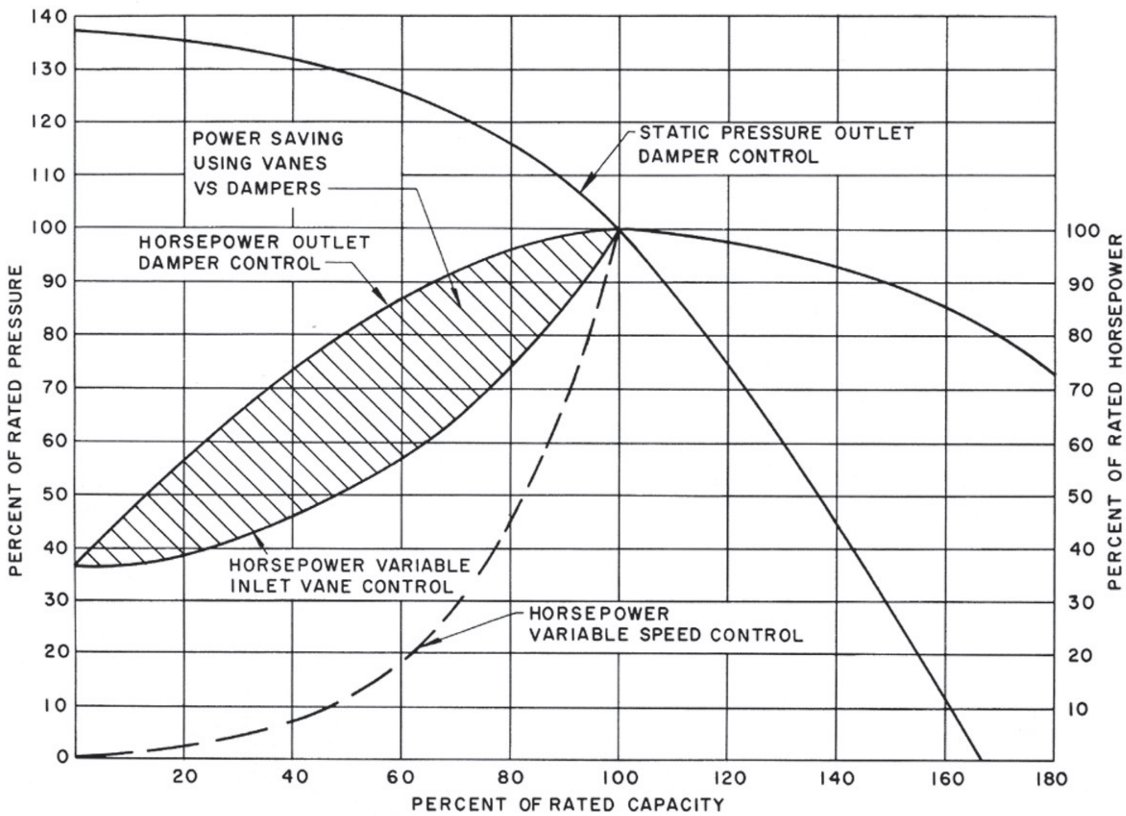
1. Variable speed motor control
2. Outlet damper control
3. Variable inlet vane control
4. Scroll volume control
5. Fan drive change

Use of a variable speed motor to control fan capacity is the most efficient means of control and the best from the standpoint of sound level. However, it is the most expensive method.

Use of outlet dampers with a constant speed motor is the least expensive method but the least expensive method but the least efficient of the first three mentioned above.

Variable inlet vanes may be used to adjust the fan delivery efficiently over a wide range. This method controls the amount of air spin at the fan inlet, thus controlling the static pressure and horsepower requirement at a given fan speed.

Figure 20 compares variable inlet vane control, outlet damper control and speed control as each affects fan performance. The horsepower curves indicate the power required at various vane settings, damper positions and fan speeds respectively.



Courtesy of Buffalo Forge Co.

Fig. 20 — Comparison of Fan Control Methods

The horsepower curve of variable inlet vane control (*Fig. 20*) is based on a fan designed with supplementary fixed air inlet vanes, such that there is no loss in efficiency when variable vanes are used instead. A loss of static efficiency as great as 10% results from the use of variable inlet vanes on a fan designed with an open inlet.

Tubeaxial and vaneaxial fans are often equipped with adjustable for matching the fan to fan to system requirements.

Propeller fans may be speed–controlled or blade–adjustable.

LOCATION

Refer to *Part 2* for the aspects of fan location. The effect of fan motor location on the system cooling load and air volume is discussed inn *Part 1.*

MULTIPLE INSTALLATIONS

Fans may be arranged in series or in parallel to provide for operating not met by the use of a single fan.

Possible series applications include:

1. Recirculating fan
2. Booster fan
3. Return air fan

A recirculating fan increases the supply air to a space without increasing the primary air (*Fig. 21*) . The purpose is to obtain greater air motion, usually in relatively lightly loaded area, or to decrease the temperature difference between supply air and room air. An industrial application prompted by the former purpose is the recirculation of air in an inspection room served by the same system as a neighboring production area.

A booster fan is used the step up the static pressure in a distribution system in order to serve a remote area, loaded intermittently; when this area is loaded, it requires a small air quantity relative to the total primary air (*Fig. 22*) . A conference room (space A) could be conditioned in this manner.

The most common series application is the return air fan, usually used on extensive duct systems to facilitate the controlling of the mixture of return air and outdoor air and to avoid the excessive room static pressures required (*Fig. 23*) . Use of a return air fan also provides a convenient method for exhausting air from a tightly constructed building.

In air conditioning, fans are seldom directly staged, with the outlet of the first being the inlet of the second. The fan efficiency and the operating economy suffers if this method is used for merely obtaining a higher static pressure.

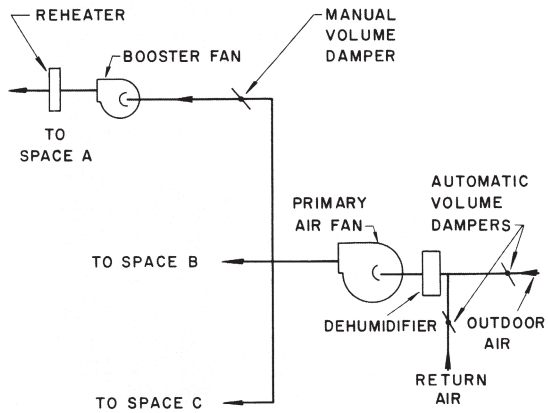


Fig. 22 – Booster Fan

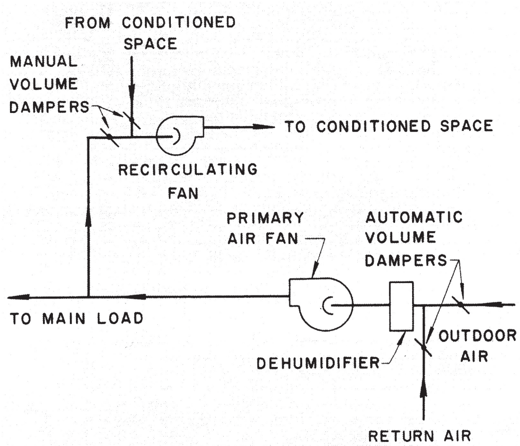


Fig. 21 – Recirculating Fan

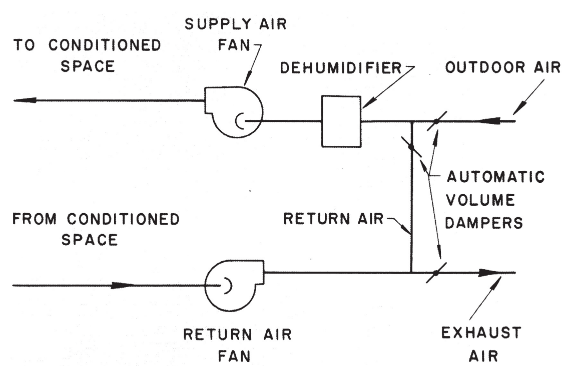


Fig. 23 – Return Air Fan

Fans may be applied in parallel if dictated by space limitations or if provision is to be made for a future addition. Centrifugal fans are available factory – mounted in cabinets for the former reason. Parallel fans provide greater capacity at a common static pressure.

However, a parallel design is seldom chosen just to increase capacity since no improvement in fan efficiency occurs and economy is not warranted by the greater first cost of the parallel installation.

CHAPTER 2. AIR CONDITIONING APPARATUS

This chapter contains practical information to guide the engineer in the application, selection and installation of various types of air conditioning apparatus, remote from the source of refrigeration.

Although the concept of air conditioning includes the moving, heating and cleaning of air, this chapter is devoted primarily to cooling, dehumidifying and humidifying equipment. Other types of air handling equipment are discussed in *Chapters 1* and *3* of this part.

TYPES OF APPARATUS

Air conditioning apparatus may be classified into two major groups:

1. Coil equipment in which the conditioning medium treats the air thru a closed heat transfer surface.
2. Washer equipment in which the conditioning medium contacts the air directly.

These two groups may be subclassified as shown *Chart 3*.

Because of its specialized application, packaged or unitary air conditioning equipment is described in *Chapter 3* of this part. Terminal equipment is discussed in *Parts 10* and *11*.

STANDARDS AND CODES

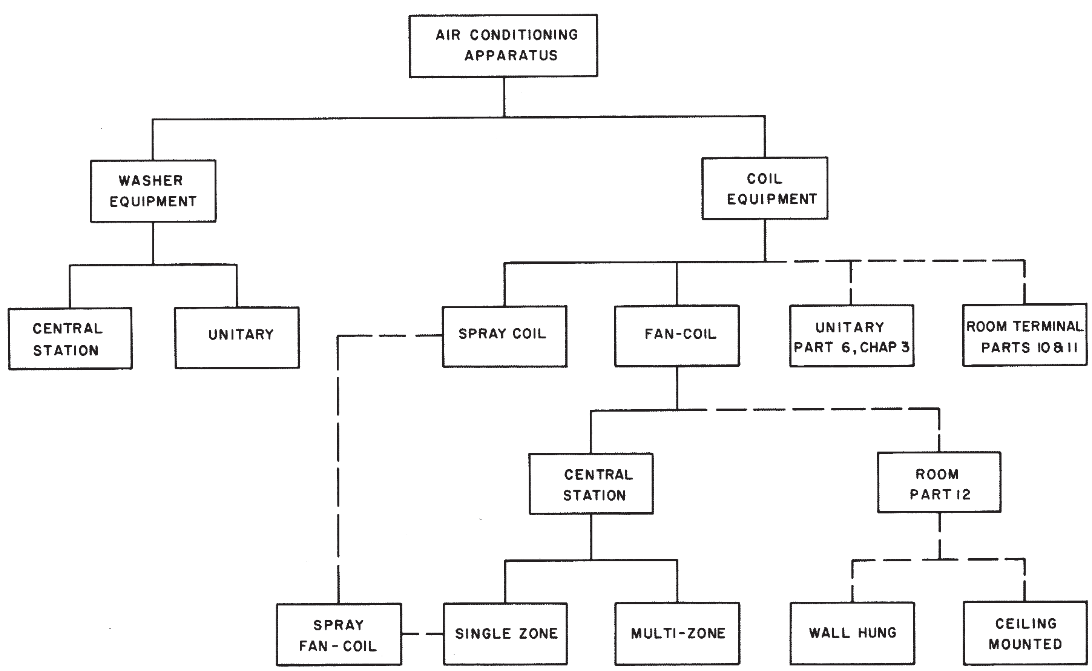
The application and installation of air conditioning apparatus should conform to all codes, laws and regulations applying at the job site.

Applicable provisions of the American Standard Safety Code B 9.1 and ARI , ASHRAE and AMCA Standards govern the testing, rating and manufacture of air conditioning apparatus.

FAN–COIL EQUIPMENT

As the term implies, the primary constituents of a fan – coil unit are a fan to produce a flow of air and a chilled water or direct expansion coil to cool and dehumidify the air.

CHART 3—APPARATUS CLASSIFICATION



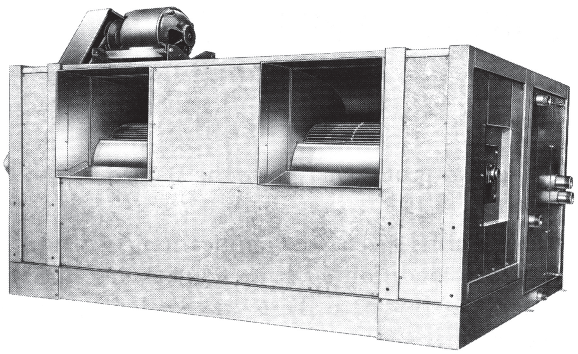


Fig. 24 – Single Zone Fan- Coil Unit

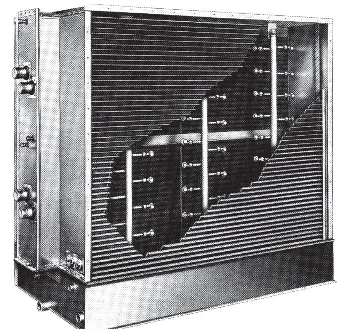


Fig. 26 – Spray Coil Section

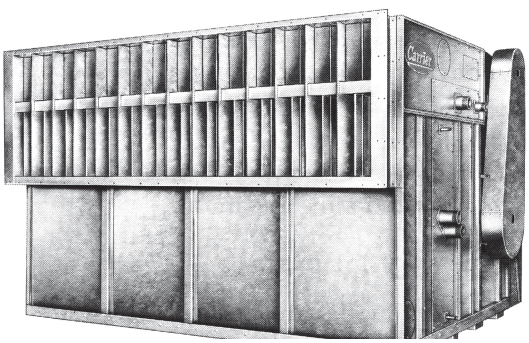


Fig. 25 – Multi-Zone Fan-Coil Unit

Accessories such as a heating coil, a humidifier and a filter section are normally available to perform, if necessary, the remaining air conditioning functions. The required components may be assembled into a factory – fabricated, cabinet style package. *Figures 24* and *25* show respectively a single zone and a multi–zone fan – coil unit.

A spray coil section is shown in *Fig. 26*. Since such equipment in intended for incorporation in a built up apparatus, it is not fan–coil equipment. However, because of the similarity of function, spray coil equipment is discussed in this section. Differences in application and layout will be noted as they exist.

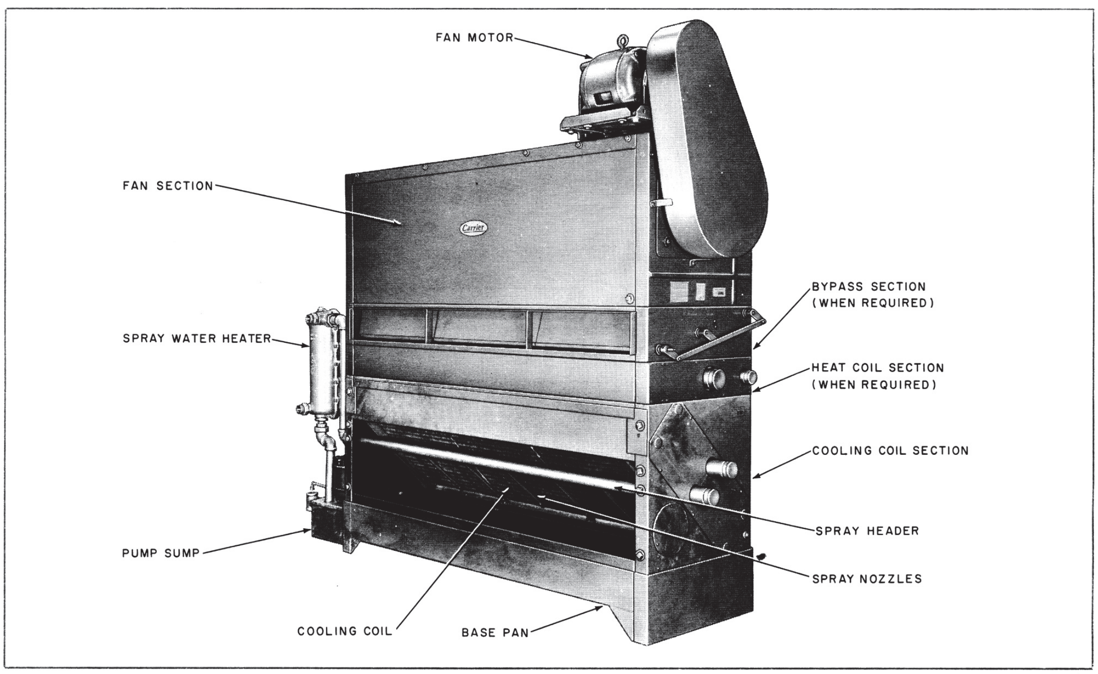


Fig. 27 — Spray Fan-Coil Unit

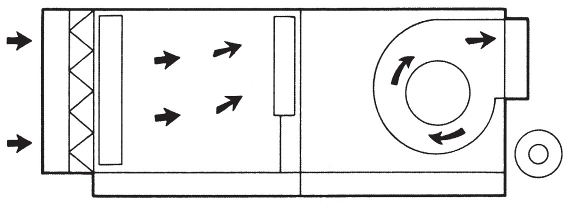


FIG. 28 — AIR FLOW — SINGLE ZONE UNIT

Figure 27 illustrates a spray fan – coil unit.

Single zone and multi–zone fan–coil units differ physically in the location of the fan relative to the cooling coil. In a single zone unit, the fan is located downstream of the cooling coil. Therefore, this type of unit is often termed a "draw–thru" unit. A multi-zone unit may be referred to as a "blow–thru" unit since the fan is located upstream of the coil . *Figures 28* and *29* indicate the flow of air thru the two types of central station fan–coil apparatus.

Typical variations occurring in total pressure, static pressure and velocity pressure, as air passes thru a fan – coil unit, are illustrated in *Fig. 30* and *31*. The use of a fan equipped with a diffuser helps to convert velocity pressure to static pressure with a minimum energy loss.

Fan–coil units are furnished with either forward or backward–curved blades. Forward-curved blade fans are well suited for such use, since they perform at slower speeds than other types of fans. Fan wheel construction is lighter in weight, more compact and less expensive than with backward–curved blades. Longer fan shafts are permissible because of the slower speeds.

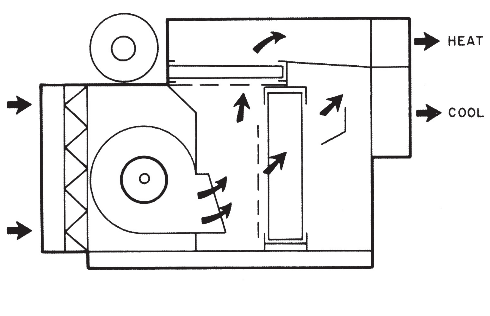


FIG. 29 — AIR FLOW — MULTI-ZONE UNIT

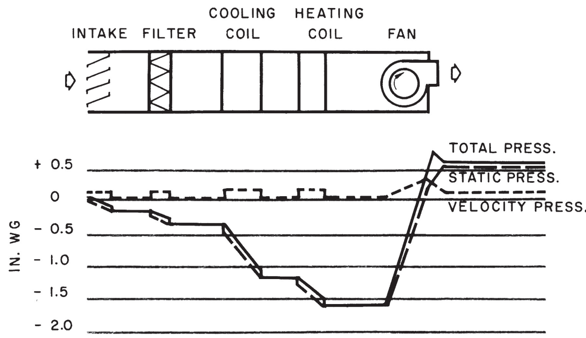


FIG. 30 — PRESSURE VARIATIONS (DRAW-THRU UNIT)

APPLICATION

The application of air conditioning equipment is influenced by the cooling load characteristics of the area to be served and the degree of temperature and humidity control required.

A single zone unit most effectively serves an area characterzied by a relatively constant or uniformly varying load. Ideally, this are would be a single large room. However, multi–room applications are practical, provided a given variation in load occurs in all rooms simultaneously and in the same proportion. If required, zoning may be accomplished by reheat or air volume control in the branch ducts.

In a multi–room application where load components vary independently and as a function of time, a multi–zone apparatus provides individual space control with a single fan unit. For this type of load a multi–zone installation is less expensive than a single zone installation with a multiplicity of duct reheat coils.

Since a multi–zone unit permits outdoor air to bypass the cooling coil at partial loads, its use is particularly adapted to applications with high sensible heat factors and minimum of outdoor air. If humidity control is required with a multi–zone unit, a precooling coil may be installed in the minimum outdoor air duct.

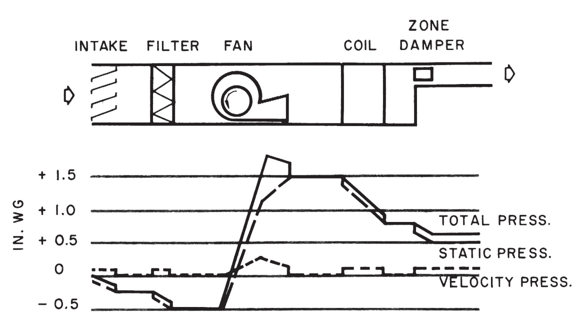


FIG. 31 — PRESSURE VARIATIONS (BLOW-THRU UNIT)

A standard fan-coil unit affords a close temperature control only. A measure of humidity control may be obtained by providing a unit humidifier such as a city water spray package. However, if more certain humidity control is required, a spray coil section or spray fan-coil unit is better suited to the application.

Spray coil equipment may be utilized for summer cooling and dehumidification, winter humidification and evaporative cooling in intermediate seasons. Its use is preferred for applications where humidity control is required, such as in industrial processes, hospitals, libraries and museums. Spray coil equipment may also be equipped with a spray water heater to provide simultaneous cooling or heating and humidification.

Standard fan-coil equipment, both single zone and multi-zone, may be obtained for air deliveries as high as 50,000 cfm. Multiple spray coil sections are available for air quantities exceeding 60,000 cfm. Where the size of available factory-fabricated equipment is exceeded the apparatus must be constructed of individual cooling coils or spray coil sections.

Static pressure limitations on fan-coil unit fans vary widely with the manufacturer considered. Available cabinet fan pressure classes are defined in *Chapter 1* of this part.

UNIT SELECTION

The selection of fan-coil equipment is a matter of achieving the required performance at the maximum economy. The economic aspect includes not only the particular unit and coil chosen but also the effect of that choice on other system components, such as piping, ductwork and refrigeration equipment.

The selection procedure involves choosing the unit size and the coil. A coil selection includes the determination of the coil depth in rows required, the 0ptimum coil fin spacing and, in the case of chilled water coils, the appropriate circuiting.

Unit Size

With the dehumidified air quantity known, the choice of unit size normally precedes the coil selection. In most cases, the size is determined by the cooling coil face velocity.

When selecting a coil face velocity, it is suggested that the highest allowable face velocity he used in the interest of economy. Manufacturers rate their (coils at maximum face velocities proven by tests to he satisfactory, with respect to both the entrainment of moisture droplets and air resistance. However, if simultaneous reheat and dehumidification is required of the unit, the maximum recommended cooling coil face velocity may be less than that otherwise allowed, depending on the design of the particular unit in question.

Since a unit reheat coil is not as deep as the cooling coil and does not condense moisture, limiting the unit size by limiting the heating coil face velocity is not economically justifiable. Manufacturers of fan-coil equipment have design their internal heating coils to provide optimum performance at recommended cooling coil face velocities.

As explained in *Chapter 1* of this part, fan outlet velocity should not be used as a selection criterion reflecting the intensity of sound generation. Sound characteristics improve with rising fan efficiency, rather than with decreased outlet velocities.

Coil Selection

A particular cooling coil is selected to produce a desired effect on the air passed thru it, in accordance with the sensible, latent and total cooling loads calculated for the space and with the condition of the air entering the coil. However, the final selection defines also the required chilled water flow, the pressure drop at that flow and the required entering water temperature; or in the case of a direct expansion coil, the refrigerant temperature. Therefore, the coil selection should be made with regard to refrigerant side or chilled water side performance as well as to air side performance.

Thus, each coil selection has two facets which may be regarded as independent for the purposes of selection. Air side and refrigerant side performance should be considered separately and then matched to produce the final economically optimum coil selection. The apparatus dewpoint method of coil selection provides means for matching air side and refrigerant side performances. This method is described in *Part 1*.

The two-step concept of coil selection is presented as follows:

1. Make a tentative coil selection in terms of rows and fin spacing, based on the bypass factor required by established air conditions. Coil bypass factor determines apparatus dewpoint.
2. Determine the refrigerant side performance, using the apparatus dewpoint found in the first step. This involves finding the required refrigerant temperature for direct expansion coils or the chilled water quantity, temperature and resulting pressure drop for water coils.

Thus, the coil can be tentatively selected without regard to the final refrigeration machine selection. If the first. coil selection does not provide satisfactory refrigerant side performance, another coil with adequate air side performance may be tried. The optimum selection assures proper performance at the least owning and operating cost.

Often in a multi-zone application, the apparatus dewpoints of the various areas differ. Rather than penalizing the cost of the entire system by selecting the lowest room apparatus dewpoint as the coil apparatus dewpoint, a higher, more representative apparatus dewpoint may be chosen, and a compromise accepted in the design relative humidity in the room with tl1e lower apparatus dewpoint. The increased relative humidity is offset by a decrease in dry-bulb temperature. Such a decision may be required in the case of a conference room, with its relatively high latent load. If a compromise is unacceptable for this application, maximum economy may be achieved by furnishing the special area with a separate system.

The various types of coil ratings and selection techniques encountered either use directly, or are derived from, one of two methods. They are the apparatus dewpoint (effective surface temperature) method and the modified basic data method. The latter involves calculating coil performance from basic heat transfer data and equations. It combines air side and refrigerant side performance determination into a single operation. However, the basic data method requires assumptions which are usually modified later in the selection, and is therefore a trial-and-error procedure. Calculated coil depth may be a decimal figure which must be rounded to a whole number, in turn necessitating a recalculation of performance. The apparatus dewpoint method is derived from the two-step concept of coil selection arid implements its use. Coil rows are dealt with in terms of standard whole numbers only.

Charts 4 and 5 are conversion charts used to evaluate the air side performance of any cooling coil, with entering and leaving air conditions established. This performance is in terms of coil bypass factor and apparatus dewpoint. A straight edge, fixed at the entering dry-bulb temperature and rotated to pass thru the various intersections of the coil bypass factor and the line connecting entering and leaving wet-bulb temperatures, indicates the coil bypass factor which satisfies the leaving dry-bulb temperature. The apparatus dewpoint can he read at the chosen intersection.

Where the bypass factor for a particular coil is unknown, the coil performance may he plotted on the chart, and the bypass factor may he read at the intersection of the entering-leaving wet-bulb and dry-bulb lines. The bypass factors of various coils may thus be directly compared.

When selecting a cooling coil in conjunction with an air conditioning load estimate form, the bypass factor of the coil selected should agree reasonably with the bypass factor assumed in the estimate. If it does not, the estimate should be adjusted accordingly, as indicated in *Part 1*.

Refrigerant side coil ratings presume a tentative coil selection when based on the apparatus dewpoint. *Chart 6* and *Table 7* illustrate apparatus dewpoint refrigerant side ratings for chilled water and direct expansion coils respectively. Such charts are used in the second step of the two-step approach described above.

Table 8 shows the entering wet-bulb type of rating for direct expansion coils. This method of presentation is used frequently and may or may not be derived from the apparatus dewpoint method.

For a direct expansion coil, optimum coil circuiting is incorporated by the manufacturer into the coil design. A direct expansion coil experiences a decreased capacity with an increased refrigerant pressure drop caused by a greater coil circuit length. This is true even with a given coil surface.

Chilled water coils are usually offered with two or more circuiting arrangements, and the final coil selection prescribes the circuiting. The coil with the least number of circuits has the greatest number of passes back and forth across the coil face and vice versa. The minimum circuited coil has a greater capacity and produces a higher chilled water temperature rise at a given water quantity.

CHART 4—CONVERSION CHART (48F TO 60F ADP)

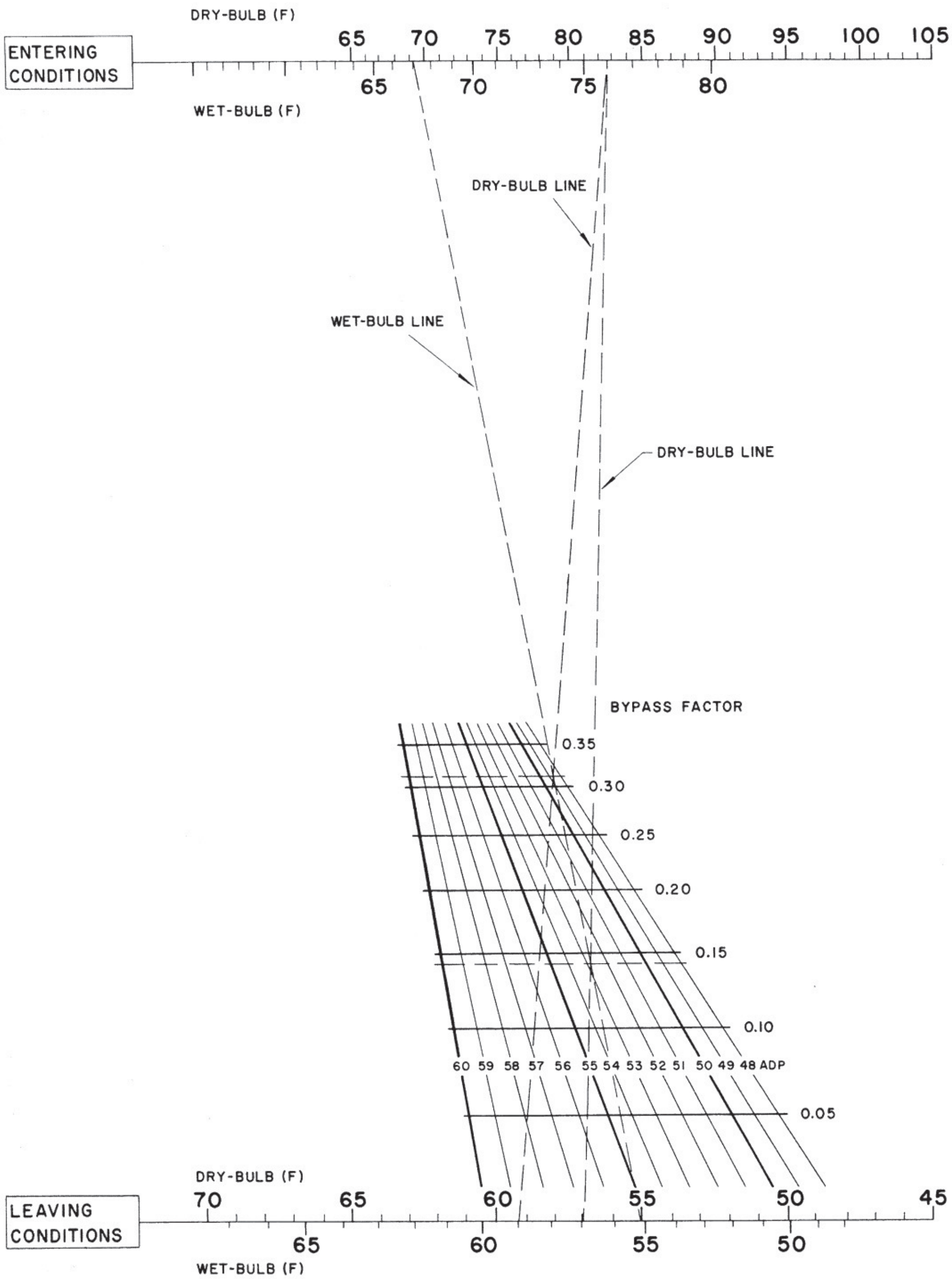


CHART 5—CONVERSION CHART (36F TO 48F ADP)

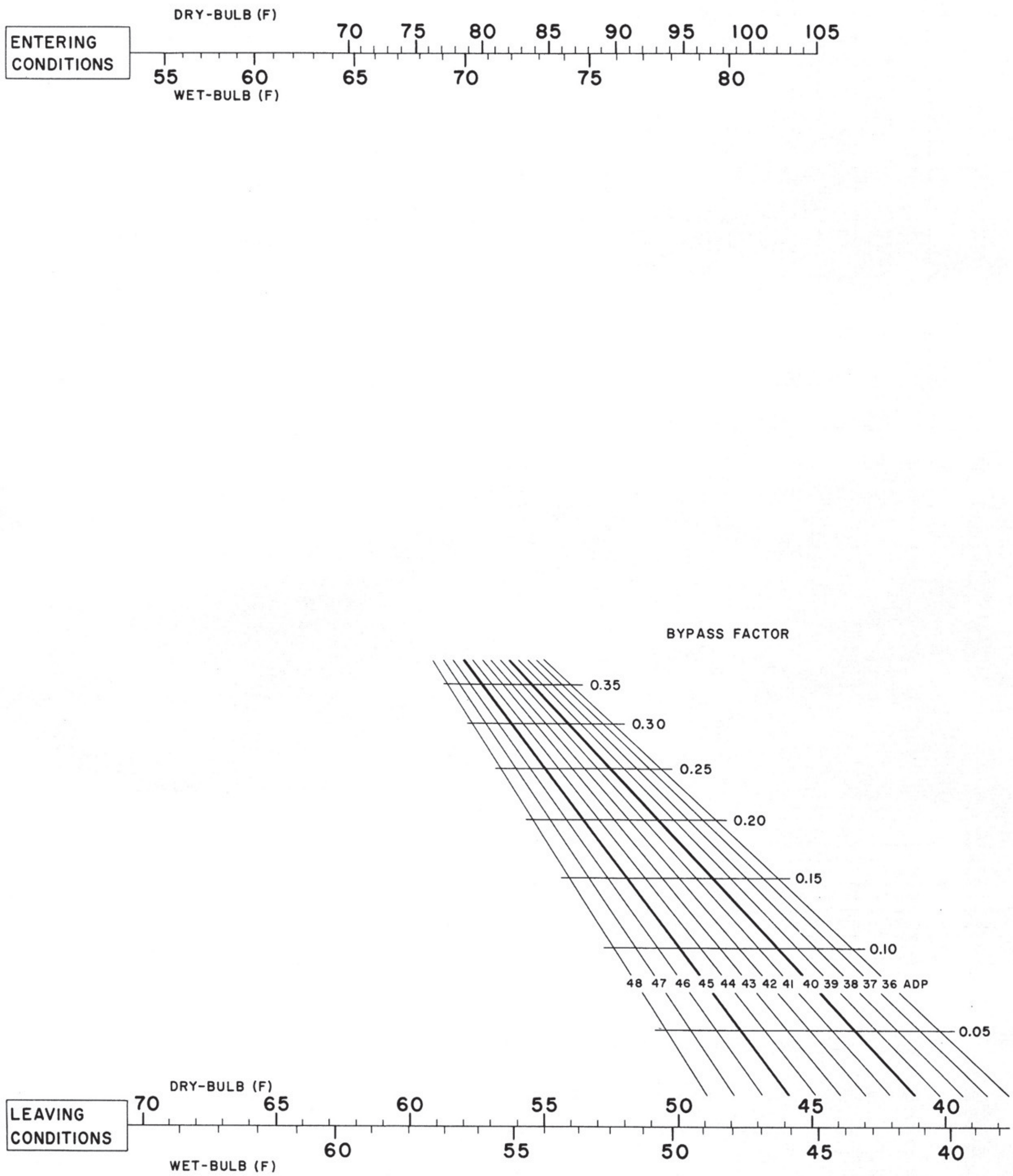


TABLE 7—DIRECT EXPANSION COIL RATINGS (APPARATUS DEWPOINT)

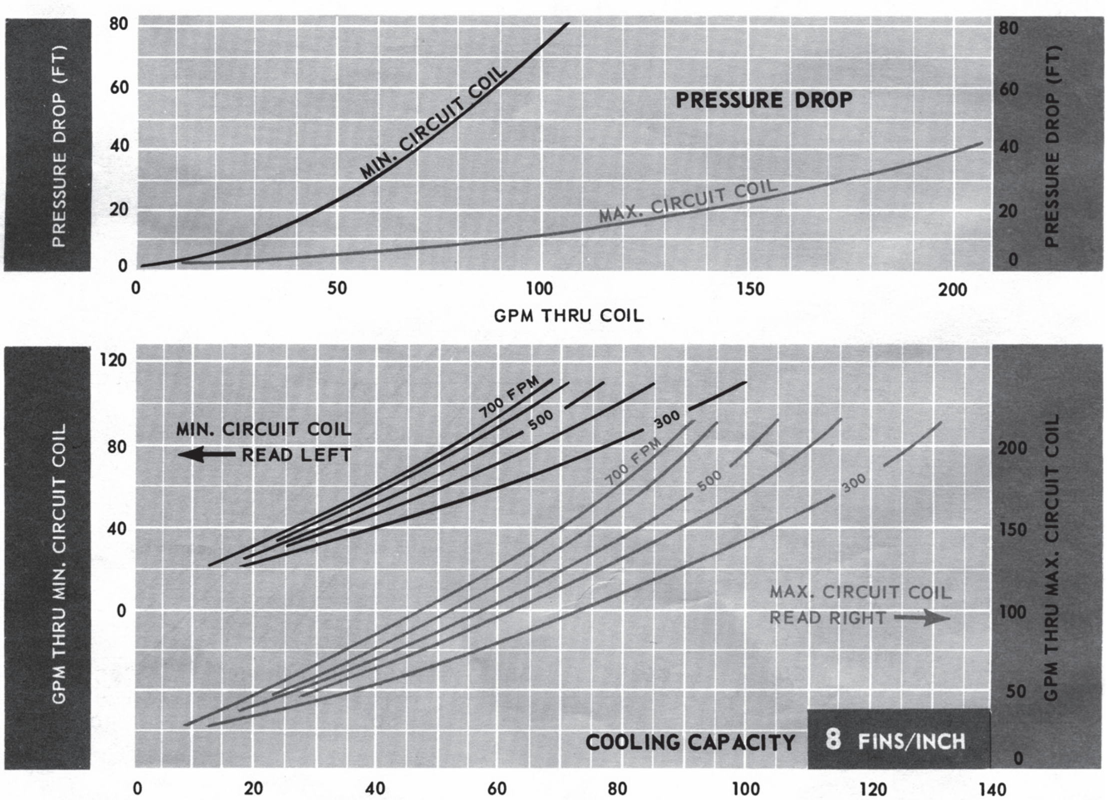
4 ROWS, 8 FINS/INCH — Refrigerant Temp (F)

GTH (1000 BTU/HR)	Apparatus Dewpoint (F) 36	38	40	42	44	46	48	50	52	54	56	58	60
48		26	28	30	32	34	36	38	40	42	44	46	48
65		26	28	30	32	34	36	38	40	42	44	46	48
82		26	28	30	32	34	36	38	40	42	44	46	48
99		26	28	30	32	34	36	38	40	42	44	46	48
115		26	28	30	32	34	36	38	40	42	44	46	48
132			26	28	30	32	35	37	39	41	43	45	47
149				26	29	31	33	35	37	39	41	43	45
166				25	27	29	31	33	35	37	39	42	44
183					25	27	29	31	33	35	37	39	42
201							26	28	31	33	35	37	39
218								26	28	30	32	35	37
235									25	28	30	32	34
252										25	27	30	32
270												26	29
288													26

4 ROWS, 14 FINS/INCH — Refrigerant Temp (F)

GTH (1000 BTU/HR)	Apparatus Dewpoint (F) 36	38	40	42	44	46	48	50	52	54	56	58	60
48	26	28	30	32	34	36	38	40	42	44	46	48	50
65	26	28	30	32	34	36	38	40	42	44	46	48	50
82	26	28	30	32	34	36	38	40	42	44	46	48	50
99	26	28	30	32	34	36	38	40	42	44	46	48	50
115	25	27	29	31	33	35	37	39	41	43	45	47	49
132	25	27	29	31	33	35	37	39	41	43	45	47	49
149		25	27	30	32	34	36	38	40	42	44	46	48
166			26	28	30	32	34	36	38	40	43	45	47
183				26	28	30	32	34	37	39	41	43	45
201					26	28	30	32	34	37	39	41	43
218						25	28	30	32	34	37	39	41
235							26	28	30	32	35	37	39
252								25	27	30	32	35	37
270									25	27	30	32	34
288										25	27	29	32

CHART 6—CHILLED WATER COIL RATINGS (APPARATUS DEWPOINT)



$$Q\ (\text{thousands}) = \frac{\text{GRAND TOTAL HEAT (BTU/HR)}}{t_{adp} - t_{ew}\ (F)}$$

TABLE 8—COOLING COIL RATINGS (ENTERING WET-BULB TEMPERATURE)

LEAVING AIR WET BULB TEMPERATURE (F) AND CAPACITY (TONS*)

Cooling Coil Face Velocity (FPM)	Refrig. Temp. (F)	Rows	Fins Per Inch	Ratings*	ENTERING AIR WET BULB TEMPERATURE (F) 65	66	67	68	69	70	71	72	73	74	75	76	77	78	79	80
600	35	4	7	LWB	53.7	54.5	55.2	55.9	56.7	57.4	58.4	59.2	59.9	60.9	61.8	62.6	63.5	64.3	65.3	66.3
				TONS	1.68	1.75	1.83	1.91	1.99	2.07	2.13	2.21	2.30	2.36	2.44	2.52	2.60	2.68	2.75	2.81
			14	LWB	50.4	51.0	51.7	52.5	53.2	54.0	54.8	55.7	56.5	57.3						
				TONS	2.12	2.21	2.29	2.38	2.46	2.55	2.63	2.72	2.80	2.89						
		6	7	LWB	50.6	51.6	52.5	53.5	54.4	55.2	56.0	56.9	57.6	58.5	59.5	60.2				
				TONS	2.08	2.13	2.19	2.25	2.32	2.39	2.47	2.55	2.64	2.71	2.78	2.87				
			14	LWB	48.5	49.3	50.2	51.0	51.8	52.6	53.6	54.5								
				TONS	2.34	2.42	2.48	2.57	2.64	2.72	2.81	2.89								
	40	4	7	LWB	55.3	56.1	56.8	57.4	58.1	59.0	59.8	60.5	61.5	62.2	63.0	63.6	64.6	65.5	66.3	67.2
				TONS	1.47	1.54	1.61	1.70	1.78	1.84	1.92	2.00	2.06	2.17	2.25	2.33	2.41	2.49	2.58	2.65
			14	LWB	52.7	53.4	53.9	54.6	55.0	55.9	56.7	57.4	58.2	59.0	59.6	60.4				
				TONS	1.82	1.91	2.00	2.09	2.18	2.28	2.37	2.46	2.56	2.64	2.75	2.84				
		6	7	LWB	53.0	53.6	54.3	55.0	55.7	56.5	57.2	58.0	58.8	59.6	60.4	61.3	62.2	63.0		
				TONS	1.78	1.87	1.95	2.04	2.12	2.21	2.29	2.38	2.46	2.55	2.63	2.72	2.80	2.89		
			14	LWB	50.6	51.3	52.0	52.7	53.4	54.1	54.8	55.7	56.5	57.3						
				TONS	2.08	2.17	2.25	2.35	2.45	2.53	2.62	2.71	2.80	2.89						
	45	4	7	LWB	57.6	58.2	58.8	59.5	60.1	60.7	61.5	62.2	63.0	63.7	64.5	65.2	66.1	66.8	67.2	68 6
				TONS	1.15	1.23	1.32	1.40	1.49	1.57	1.65	1.74	1.82	1.90	1.99	2.07	2.15	2.24	2.31	2.38
			14	LWB	54.8	55.5	56.1	56.8	57.5	58.1	59.0	59.6	60.2	60.9	61.5	62.2	62.8	63.8	64.6	
				TONS	1.53	1.61	1.70	1.78	1.87	1.96	2.05	2.15	2.25	2.36	2.47	2.59	2.76	2.85	2.97	
		6	7	LWB	54.8	55.5	56.1	56.8	57.5	58.2	59.0	59.7	60.5	61.3	62.2	62.8	63.5	64.4	65.2	66.0
				TONS	1.53	1.61	1.70	1.78	1.87	1.96	2.04	2.13	2.21	2.30	2.38	2.47	2.58	2.66	2.76	2.85
			14	LWB	52.7	53.4	54.1	54.7	55.4	56.0	56.6	57.5	58.3	58.9	59.6	60.0				
				TONS	1.81	1.90	1.99	2.08	2.17	2.27	2.38	2.45	2.55	2.65	2.76	2.89				
	50	4	7	LWB	60.5	61.1	61.5	62.0	62.5	63.2	63.7	64.2	65.0	65.6	66.4	66.9	67.6	68.5	69.2	70.0
				TONS	0.72	0.81	0.92	1.02	1.12	1.20	1.32	1.41	1.50	1.59	1.67	1.78	1.87	1.95	2.06	2.12
			14	LWB	57.9	58.3	58.8	59.4	59.8	60.3	61.0	61.8	62.4	63.0	63.6	64.1	65.0	65.8	66.5	67.2
				TONS	1.10	1.22	1.32	1.42	1.53	1.64	1.73	1.83	1.93	2.03	2.12	2.24	2.34	2.43	2.55	2.64
		6	7	LWB	57.6	58.2	58.7	59.3	59.8	60.4	61.0	61.7	62.5	63.0	63.7	64.5	65.2	65.9	66.6	67.2
				TONS	1.13	1.23	1.33	1.43	1.53	1.62	1.72	1.83	1.91	2.01	2.12	2.21	2.30	2.40	2.51	2.64
			14	LWB	55.6	56.1	56.6	57.1	57.6	58.2	58.9	59.5	60.1	60.8	61.4	62.1	62.9	63.6	64.2	
				TONS	1.42	1.52	1.63	1.74	1.84	1.95	2.05	2.16	2.27	2.38	2.49	2.59	2.69	2.80	2.92	

LEAVING AIR TEMPERATURE DIFFERENCE (F) (LEAVING DRY BULB MINUS LEAVING WET BULB) (LDB-LWB)

ENTERING AIR TEMPERATURE DIFFERENCE (F) (ENTERING DRY BULB MINUS ENTERING WET BULB) (EDB-EWB)

Rows	Fins Per Inch	Refrig. Temp. (F)	600 FPM Coil Face Velocity 10	11	12	13	14	15	16	17	18	19	20	700 FPM Coil Face Velocity 10	11	12	13	14	15	16	17	18	19	20
4	7	35	2.1	2.4	2.7	3.0	3.3	3.6	3.9	4.2	4.5	4.7	5.1	2.6	2.9	3.2	3.5	3.8	4.1	4.4	4.8	5.1	5.4	5.7
		40	2.3	2.6	2.8	3.2	3.5	3.7	4.0	4.3	4.6	4.9	5.3	2.7	3.0	3.3	3.6	3.9	4.2	4.5	4.9	5.2	5.5	5.8
		45	2.6	2.9	3.2	3.4	3.7	4.0	4.3	4.6	4.9	5.2	5.5	2.8	3.1	3.4	3.7	4.0	4.4	4.6	5.0	5.4	5.6	6.0
		50	2.7	3.0	3.3	3.6	3.9	4.2	4.5	4.7	5.0	5.4	5.7	3.0	3.3	3.7	4.1	4.4	4.8	5.1	5.5	5.8	6.2	6.5
	14	35	1.0	1.1	1.2	1.3	1.5	1.6	1.7	1.9	2.0	2.1	2.2	1.0	1.1	1.3	1.5	1.6	1.7	1.9	2.0	2.1	2.3	2.4
		40	1.0	1.1	1.2	1.3	1.5	1.6	1.7	1.9	2.0	2.1	2.2	1.3	1.4	1.6	1.7	1.9	2.0	2.1	2.2	2.4	2.5	2.6
		45	1.1	1.2	1.3	1.5	1.6	1.7	1.8	1.9	2.0	2.1	2.2	1.8	1.9	2.0	2.2	2.3	2.4	2.6	2.7	2.9	3.0	3.2
		50	1.1	1.3	1.4	1.5	1.6	1.7	1.8	1.9	2.1	2.2	2.3	2.0	2.1	2.2	2.4	2.6	2.8	2.9	3.0	3.1	3.3	3.5
6	7	35	1.2	1.3	1.4	1.6	1.8	1.9	2.1	2.3	2.4	2.5	2.7	1.4	1.6	1.9	2.0	2.2	2.3	2.4	2.6	2.8	3.0	3.2
		40	1.2	1.3	1.5	1.7	1.8	1.9	2.1	2.4	2.5	2.6	2.8	1.7	1.9	2.1	2.2	2.4	2.6	2.8	3.0	3.1	3.3	3.5
		45	1.4	1.5	1.7	1.8	1.9	2.1	2.3	2.5	2.6	2.7	2.9	2.0	2.1	2.3	2.4	2.7	2.8	3.0	3.2	3.4	3.6	3.8
		50	1.5	1.6	1.8	1.9	2.1	2.2	2.4	2.5	2.7	2.8	3.0	2.0	2.3	2.5	2.6	2.8	3.0	3.2	3.4	3.6	3.7	3.9
	14	35	0.3	0.4	0.5	0.5	0.6	0.6	0.6	0.6	0.7	0.7	0.8	0.2	0.2	0.3	0.3	0.4	0.5	0.5	0.5	0.6	0.7	0.8
		40	0.3	0.4	0.5	0.5	0.6	0.6	0.6	0.6	0.7	0.7	0.8	0.3	0.3	0.3	0.4	0.5	0.5	0.6	0.6	0.7	0.7	0.8
		45	0.4	0.4	0.5	0.5	0.6	0.6	0.6	0.6	0.7	0.7	0.8	0.7	0.8	0.8	0.9	0.9	0.9	1.0	1.0	1.1	1.1	1.2
		50	0.4	0.4	0.5	0.5	0.6	0.6	0.6	0.6	0.7	0.7	0.8	1.2	1.2	1.3	1.3	1.4	1.4	1.5	1.5	1.6	1.7	1.9

*Tons given per square foot of coil face area.

However, the greater number of passes of a minimum circuited coil results in a pressure drop higher than that thru a coil of the same size but with more circuits and less passes. Minimum circuited coils are often used on large extensive systems in which the greater pumping head required is more than offset economically by the reduced first cost of piping and insulation.

With the required air side coil performance given, the greater the difference between apparatus dew-point and entering chilled water temperature the smaller the required water quantity will be. Therefore, the choice of a chilled water temperature may involve an economic analysis of the first costs and operating costs of the refrigeration plant versus the cost of the piping system. The selection of the water temperature should not be arbitrary; however, experience has shown that a temperature approximately 5 degrees below the apparatus dewpoint is the maximum water temperature that should be used to effect an economical system design. If the resulting water quantities seem to be too high , a lower temperature can be assumed, and its influence on the refrigeration machine size, power input and piping costs should be studied. With a given coil, load and apparatus dewpoint, when the chilled water temperature is reduced , the required water quantity decreases and the temperature rise increases.

By using a coil which requires a smaller water quantity at a higher temperature rise, the following advantages may be realized:

1. a. A smaller refrigeration machine may be selected, or
   b. The horsepower requirement may be reduced for the same size machine by operating at an increased evaporator temperature, *or*
   c. The condenser piping or heat rejection equipment may be reduced for the same size machine by operating at a higher condensing temperature with less condenser water.
2. Lower chilled water distribution costs with saving in piping, pump and insulation may be obtained.

A limitation is also imposed on the minimum chilled water quantity by the velocity required for efficient heat transfer. A minimum Reynolds number of 3500 is suggested toinsure predictable and efficient performance of a coil. The minimum chilled water flow required to maintain this Reynolds number is approximately 0.9 gpm per circuit for a $^5/_8$ in. OD coil tube diameter. For a $^1/_2$ in OD tube diameter, the minimum flow suggested is 0.7 gpm per circuit.

Well water may be circulated thru chilled water coils if it is sufficient quantity and at a satisfactory temperature. However, well water temperature are usually low enough to produce sensible cooling only and little or no latent heat removal. In such a case, the well water may be utilized in a precooling coil to remove some of the sensible heat. The remaining cooling load ,sensible and latent , is handled by supplement refrigeration.

Manufacturer's recommendations regarding maximum and minimum direct expansion coil loadings should be followed. Selections at loadings below the minimum may result in unsatisfactory oil return, poor refrigerant distribution and coil frosting.

Atmospheric Corrections

Cooling coil ratings are based upon the standard atmospheric conditions of 29.92 in. Hg barometric pressure. For atmospheric pressures significantly different, such as at altitudes exceeding 2500 feet, a correction should be applied to the air quantity before making the coil selection.

Assuming that the necessary corrections have been made to the load calculation and the sensible heat factor as described in *Part 1,* the following procedure should be applied to the unit selection:

1. Obtain the density ratio from *Chapter 1, Chart 2.*
2. Multiply the calculated dehumidified air quantity by the density ratio to determine the equivalent air flow at sea level.
3. Use this adjusted air quantity, together with the calculated cooling load and sea level refrigerant side coil ratings, to determine the coil water flow and pressure drop or refrigerant temperature.

The calculated dehumidified air quantity is used with no correction to determine unit size and coil face velocity. However, the coil air side pressure drop must he corrected, as described in *Part 2.*

Fan performance is analyzed in *Chapter 1* of this part and motor selection is influenced as outlined in *Part 8.*

ACCESSORIES

Heating Coils

Unit heating coils for fan-coil equipment are available in a variety of depth and fin spacing combinations and in both the nonfreeze steam and U-bend types. The latter type may be used with hot water or steam and may be obtained in different combinations of tube face and fin spacing to produce different rises with the same entering air temperature, face velocity, and steam pressure or hot water temperature. Heating coils are also usually capable of being mounted before or after the cooling coil.

During the intermediate season, any zone served by a multi-zone unit should be able to obtain heating or cooling on demand. Since there is no common supply air duct in which air mixing can occur on a multi-zone installation, the problem of air temperature stratification is of considerable importance. Stratification across the unit heating coil may cause some zones to be denied heat when it is needed.

The throttling of a steam control valve may produce stratification if the steam condenses fully before reaching the end of the tube or circuit. Therefore, it is suggested that full steam pressure he applied to a multi-zone unit heating coil whenever heating may be required by any zone.

In order to provide an air path of approximately equal pressure drop thin either of the two widely differing beat transfer surfaces in a multi-zone fan-coil unit, perforated balancing plates are often used. It may be necessary for the engineer to select such a device, particularly if no heating coil is required.

The application and selection of heating coils is described in detail in *Chapter 4* of this part.

Humidifiers

On a fan-coil unit not equipped with recirculated water sprays, humidification may be obtained by means of a city water spray humidifier, a steam pan humidifier, a steam grid humidifier or a humidifying pack. Spray coil and steam grid equipment provides the most effective humidity control.

A city water spray humidifier consists of a header, spray nozzles and strainer. Either atomizing or non-atomizing sprays are available. The latter type requires a lower water pressure. In either case, the spray density or amount of water circulated per square foot of cooling coil face area is considerably less than that of a recirculated spray coil. Therefore, although lower in first cost, the city water spray humidifier is less efficient than a recirculated spray coil. An eliminator is not usually required for a city water spray.

The use of copper fins on copper tubes with spray humidifiers is suggested when city water has a specific electrical conductance of 500 or more micromhos at 77 F. (See *Part 5* for values of water conductance in various locations.) Aluminum fins may be used if city water is of the proper quality. The use of copper fins should be

considered where industrial gases such as hydrogen sulphide, sulphur dioxide or carbon dioxide are present and where salty atmospheres prevail.

If air flow is not maintained thru a spray section when it is operating, wetting of the unit and leakage may result. Therefore, a solenoid valve should be installed or other suitable precautions taken to stop the sprays when the unit fan is not running. To maintain a minimum coil air flow when utilizing face and bypass dampers, a minimum closure device should be provided on the face dampers.

The spraying of heating coils may result in scaling on the coil and the production of odors. This practice should therefore be avoided.

The use of spray humidifiers with a multi-zone unit should be avoided. Since the coil is subjected to a positive static pressure, spray water may leak from the unit cabinet. If sprays are used, they should be of the atomizing type. A grid or pan humidifier is preferred for this usage.

Grid humidifiers are lengths of perforated steam piping wrapped with wicking such as asbestos. The pipe is mounted in an open pan, pitched to facilitate condensate drainage. The condensate drain line from the unit should be trapped to provide a water seal, as described in *Part 3.* Steam pressures should not exceed 5 psig for this application, and the steam used should be free of odors.

The mixing of steam with conditioned air normally produces a negligible increase in the air dry-bulb temperature. This type of humidification, therefore, approximates a vertical line on a psychrometric chart. *Figure 32* illustrates the process. When designing a system using a grid humidifier, the temperature of the air entering the humidifier should be high enough to permit a moisture content at saturation (point *C)* equal to or greater than the desired air moisture content.

Pan humidifiers include a pan to hold water, a steam coil to evaporate the water, and a float valve for water make-up. For this application, a steam pressure of 20 psig is suggested for maximum humidifying efficiency.

Humidifying packs use a fill (often of glass fibers) as an evaporative surface. The pack is located in the air stream and water is sprayed over the fill.

Spray Water Heaters

Spray coil equipment may be provided with a spray water heater to permit simultaneous cooling or heating and humidification. Such flexibility is required during winter operation or where the volume of outdoor air is large in relation to the total air quantity. Typical applications include certain industrial processes or hospital operating rooms. These processes are described in *Part 1.*

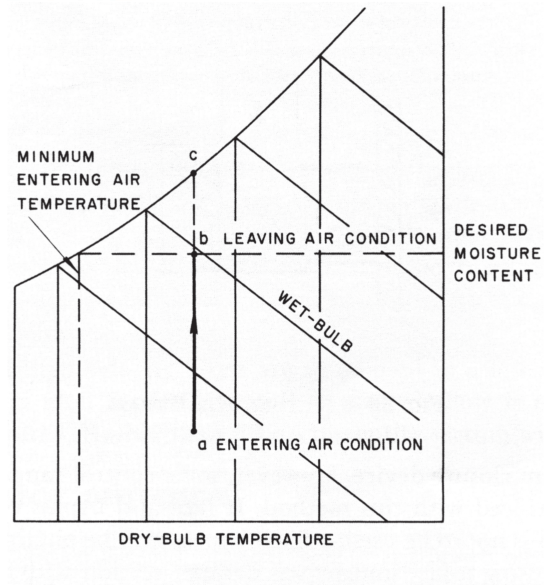


Fig. 32 — Humidification With Steam

Face and Bypass Dampers

On applications employing face and bypass control of coil equipment, the fan selection anti air distribution system should be based on an air quantity 10% above the design dehumidified air volume. This additional air quantity compensates for leakage thru a fully closed bypass damper and for air quantity variations occurring when face and bypass dampers are in an intermediate position. With the bypass dampers fully open, system static pressure may be reduced and air quantity and fan brake horsepower increased. Therefore, on face and by applications especially, fan motors should be selected so that nominal horsepower ratings are not exceeded.

Where a fixed bypassed air quantity is required, the bypass damper may be provided with a minimum closure device. However, some control range is sacrificed with this method. If face and bypass control is not to be used, a fixed bypass may be obtained by using a face and bypass damper section with the face dampers removed.

The bypass of outside and return air mixtures introduces high humidity air directly to the conditioned space. When employing face and bypass control, it is preferable to bypass return air only. This may be accomplished as shown in *Fig. 33.*

Vibration Isolation

Four types of isolators are normally used to absorb the vibrations produced by fan-coil equipment as well as other types of rotating or reciprocating machinery. In order of decreasing effectiveness and first cost, they are:

1. Steel coil springs
2. Double rubber-in-shear

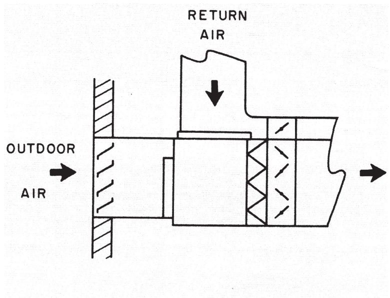


Fig. 33 – Return Air Bypass

3. Single rubber-in-shear
4. Cork

Steel spring or rubber-in-shear isolators are available for floor-mounted or suspended equipment. Ribbed neoprene pads may be bonded to any of the isolators noted above for floor-mounted units. These pads resist horizontal movement, compensate for slight irregularities in the floor surface, and protect floors from marring.

The proper bearing surface should be provided for cork pad isolators as recommended by the isolator manufacturer. Underloading does not permit the full resiliency to be utilized, while overloading may result in permanent deformation of the cork structure.

Similarly, if spring or rubber isolators are loaded past the point of full compression, binding occurs and there is no isolation.

Vibration isolation efficiency is the percentage of a vibration of a given frequency absorbed by the isolator. Thus, the vibration transmitted beyond the isolator is the difference between 100% and the isolation efficiency.

Isolation efficiency is a function of the isolator deflection when loaded and the disturbing frequency of the machine isolated. For a fan or fan-coil unit the disturbing frequency is the fan speed. *Chart 7* shows the relation between static deflection, disturbing frequency and isolation efficiency for any case of vibration. In addition, *Chart 7* indicates the ranges of deflection for which the various types of isolators are normally obtainable.

As illustrated by *Chart* 7, for a given disturbing frequency, isolator efficiency increases with deflection. Since greater deflections are obtainable with springs than with other types of isolators, springs provide the most effective isolation over all frequencies. Cork is not an effective isolation material for frequencies below about 3000 rpm.

A minimum vibration isolation efficiency of 85% is usually satisfactory for ground floor or basement applications in noncritical buildings. Upper floors may require as high an efficiency as 93%, while critical upper floors usually allow no less than 95%. When several pieces of vibrating equipment are concentrated in one room in a critical upper floor installation, the required efficiency may approach 98.5%, and the transmission of vibration to the floor should be considered in the building design.

Whether floor-mounted or suspended, unit may be mounted on a steel channel base which is then isolated. Units may also he mounted on vibration isolators directly, with no intermediate base. Manufacturers provide support points or hanging brackets for fan-coil equipment, and their recommendations regarding support points and isolator loadings should be followed. Often, larger units of a series or those including the most components require a channel frame base for mounting. Mixing boxes and low velocity filter boxes are usually mounted on their own isolators, so as to prevent a cantilever effect.

If a unit is to be directly isolated with no base employed, the deflection at each Support point should be the same. If equal loading is assumed, individual isolators may be overloaded to the point of binding or underloaded, resulting in decreased isolation efficiency. Point loadings for a given unit, coil and component are usually available from the unit manufacturer.

Operating weights should be used in selecting vibration isolators. This is particularly important where water coils are used.

Example 4 – Vibration Isolator Selection

Given:

Fan – coil unit operation weight – 1640 lb equally distributed at four points.
Fan speed - 800 rpm
Design isolation efficiency – 90 %

Find:

Required type and characteristics of isolator.

Solution:

1. From Chart 7 read the required isolator deflection of 0.6 in., within the spring application range.
2. Determine the individual isolator loading; $\frac{1640}{4}$ = 410 lb.
3. Select a spring isolator with a maximum characteristic of 410/0.6 or 683 lb per inch. If the characteristic of the spring chosen is less, the deflection is greater than 0.6 in., and the isolation efficiency greater than 90% . However, the spring must not be loaded above its maximum.

Filters

Factory–fabricated filter sections for both high velocity or low velocity filters are normally obtainable from the manufacturer of a fan–coil unit. Either throw–away or

cleanable filters can be used. For built – up apparatus field – assembled filter frames are available.

If high velocity filters are to be used in a low velocity filter section, the full area of air flow is not required. Rather than fill up the entire section with high velocity filters operating at a low velocity, bank–off pieces may be installed, thus lowering the effective area. Blank-offs should be located uniformly across the face of the filter section instead of concentrated in one place.

Filters are discussed in detail in *Chapter -4* of this part.

INSTALLATION

Location

The economic and sound level considerations pertaining to the location of air handling apparatus, as discussed in *Part 2,* are applicable to fan-coil equipment.

Two of the most important factors in the location of air conditioning equipment are the availability of outdoor air and the ease of air return. Outdoor air may be brought to a unit thru a wall, roof or central building chase. It is preferable to locate outdoor air intakes so that they do not face walls of spaces where noise would be objectionable. Air may be returned thru a duct system or directly to the equipment room.

Layout

A fan-coil unit may be of the vertical or horizontal type, depending upon the direction of air flow entering the fan cabinet. It may be floor-mounted or, in the case of a horizontal unit, suspended from above. The choice of unit style and mounting usually depends upon space requirements and optimum duct layout. A support base may be employed, if necessary, as discussed under *Vibration Isolation.*

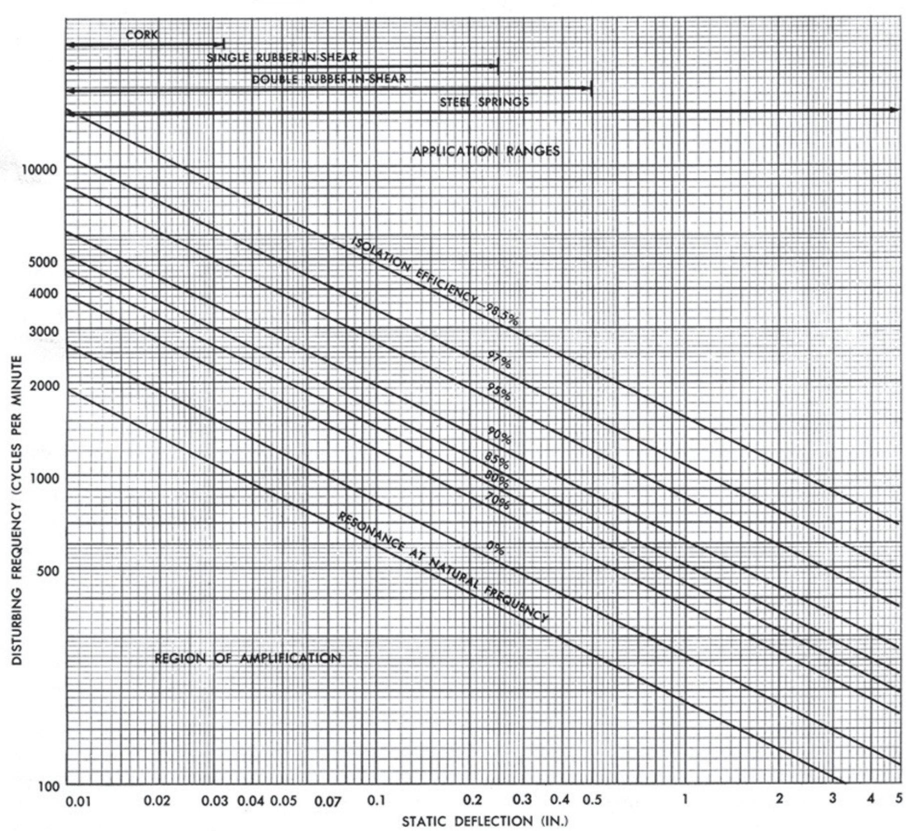
A practical location recognizes the need for effective servicing of the unit. A minimum of 30 inches is suggested to provide access between the unit and the nearest wall. This facilitates servicing of steam traps, fan bearings, damper motors and fan motor. In addition, service space about the unit must be provided for filter removal, coil removal, fan shaft removal, an(l the cleaning of cleanable coils.

Suspended units should be accessible from above, if possible. If frequent access is required and space permits, a catwalk may be required.

An access plenum and door should be provided between the filter section and coil section of a spray fan-coil unit. This access permits periodic inspection and cleaning of the sprays and drain pan.

A level unit is necessary to insure proper drainage from coils and drain pan. Manufacturers of spring

CHART 7—VIBRATION ISOLATOR DEFLECTION



vibration isolators usually provide leveling devices in the isolator to compensate for deflection differences.

Units located outdoors require suitable motors and the protection of fan drive and shaft bearings, as well as insulation as noted below.

For information pertaining to the design of air distribution system components and piping at the unit, refer to *Parts 2 and 3.*

Insulation

In a fan-coil unit the casing housing the fan section, cooling coil section and components downstream of the cooling coil are usually internally insulated. This insulation is adequate for normal interior applications. The outdoor air intake duct should be insulated and vapor sealed to prevent condensation on the duct during cold weather.
If the intake is kept as short as possible, insulation costs are minimized. Insulation and vapor sealing of the mixing box may be required, depending on the quantity of outdoor air introduced and on the winter design temperature. Intakes for units circulating 100% outdoor air should be insulated up to the preheater.

Units located outdoors should be completely covered and caulked with weatherproofing material. If the outdoor air temperature can fall below the dewpoint of the air within the unit, the unit should be externally insulated, vapor sealed and weather-proofed to prevent interior condensation and to minimize heat losses. The insulation on the top surfaces of the unit should be slightly crowned so that water can run off.

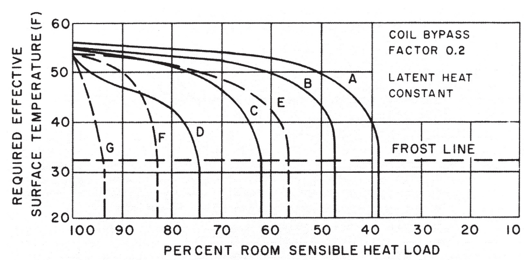
CONTROL

If an air conditioning apparatus is to perform satisfactorily under a partial load in the conditioned area, a means of effecting a capacity reduction in proportion to the instantaneous load is required. The three methods most commonly employed for capacity control are air bypass control, chilled water control and air volume control.

With a drop in room load the sensible heat ratio usually decreases since the room latent load remains constant. This condition commonly occurs in areas where a large proportion of sensible load such as solar heat gain may be decreased with no influence on latent load, as from people or infiltration.

In order to maintain design conditions at partial loads and with decreased sensible heat ratios, the effective coil surface temperature for a given coil must be lower than that surface temperature consistent with full load design conditions. This requirement is illustrated in *Fig. 34* . The relation between effective coil surface temperature and percent of design room sensible heat depends on the volume of outdoor air conditioned by the coil.

However, decreasing the flow of chilled water thru the coil as a means of capacity control causes the effective coil surface temperature to rise as the load decreases. Therefore, room humidity also rises. For this reason it is preferable to maintain the design flow of chilled water thru the coil at all times.



A—no outdoor air

15% Outdoor Air	35% Outdoor Air
B—return air bypass	E—return air bypass
C—mixture air bypass	F—mixture air bypass
D—outdoor air bypass	G—outdoor air bypass

Fig. 34 — Required Coil Performance

Figure 35 a shows a typical cooling coil process at full load for a given set of entering air and water conditions. Figures 35 b, 35 c and 35 d depict at half load the three methods of control cited and the influence in each case on effective coil surface temperature. Air volume control is similar in effect to air bypass control. However, the bypassing of air around the coil permits a relatively constant air delivery to be maintained.

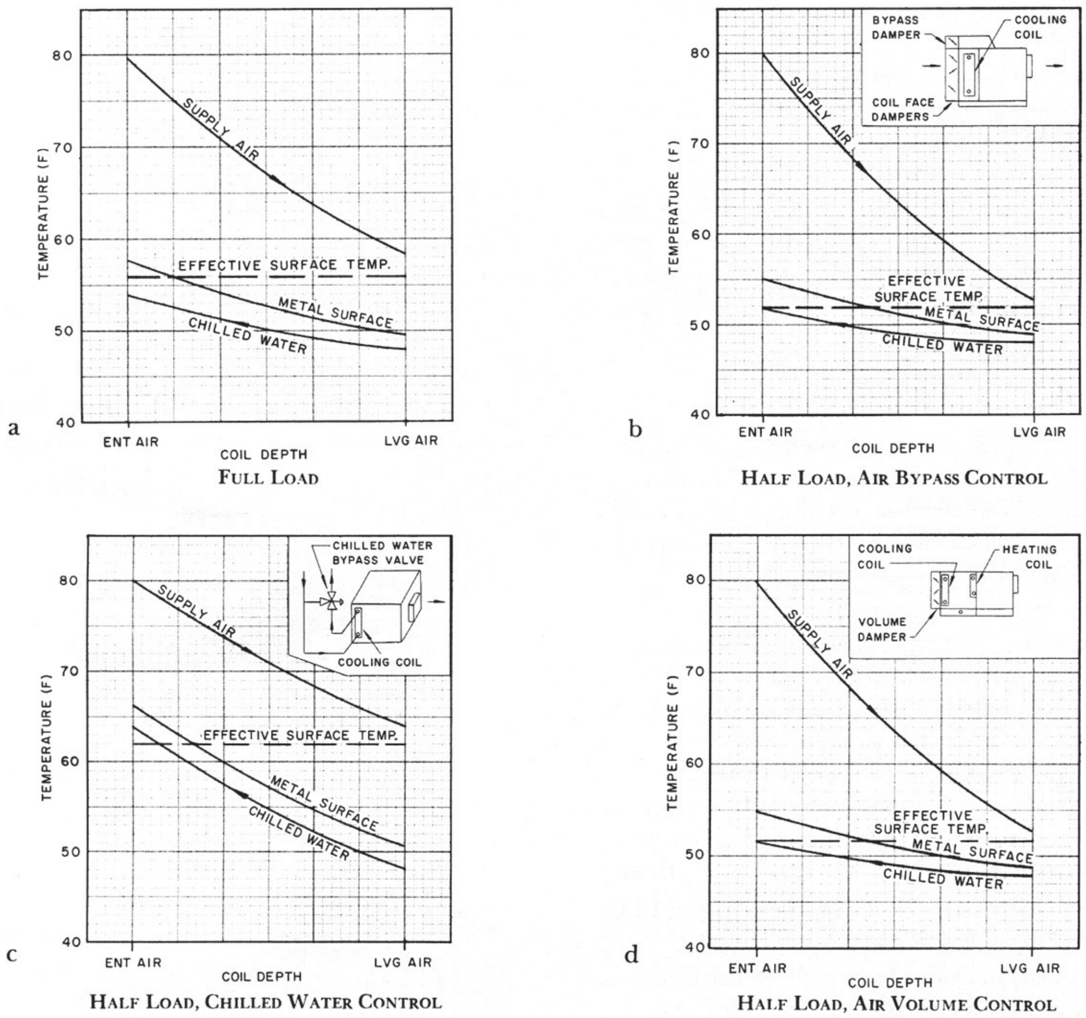


FIG. 35 — TYPICAL COOLING COIL PROCESSES

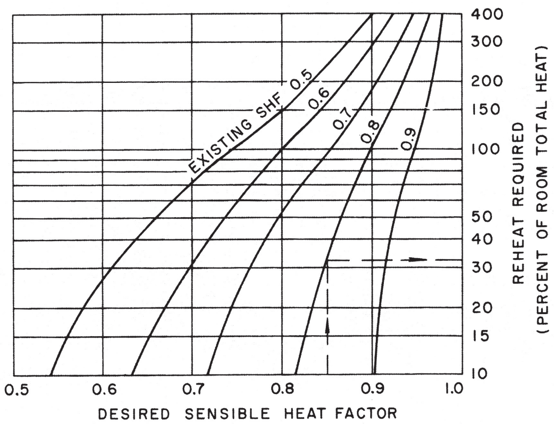


Fig. 36 — Reheat Control Requirements

As discussed in Part 1, applications with high latent loads may require reheat control of room temperature. *Figure 36* indicates the amount of reheat required, relative to room total heat, to maintain design relative humidity at various sensible heat ratios.

COIL FREEZE-UP PROTECTION

The freezing of water in preheat, reheat and chilled water coils may damage the coils and lead to costly repairs. Freezing may occur not only in coils of units operating during cold weather but also in the coils of units not in operation.

Outdoor air at subfreezing temperatures often comes in contact with heat transfer surfaces as a result of air temperature stratification. Stratification is caused usually by incomplete mixing of return air and outdoor air or by an uneven temperature rise thin the preheat coil caused by coil throttling. The complete mixing of air may be promoted by the proper arrangement and design of the ductwork. Uneven temperature rises thru preheat coils and heating coil freeze-up may be prevented as outlined in *Chapter -1* of this part.

Coil freeze-up may also be caused by the direct introduction of cold air thru an unprotected coil. Circulation of outdoor air thru an interior fan-coil unit not in operation can be induced by a stack effect, particularly if the unit is on one of the lower floors of a tall building.

In addition to design precautions against stratification, the following methods may be employed to protect a water coil:

1. Remove the water from the coil during the winter.
2. Run the chilled water pump.
3. Decrease the freezing point of the coil water.

Removal of the water from the coil should be accompanied by blowing out the coil with a portable blower to remove residual water. An alternate method of freeze protection is to circulate an inhibited antifreeze solution thru the coil before final drainage.

Operating the chilled water pumped during the winter is a costly solution to the problem of freezing. In addition, it is not a certain method since a plugged tube could still freeze.

The practice of using a properly inhibited alcohol brine or antifreeze thruout the year for coil freeze-up protection is becoming more common. Brines have been developed and are now available, particularly for this purpose. Refer to *Part 4.*

WASHER EQUIPMENT

The most commonly applied type of washer equipment is the central station washer *(Fig. 37),* designed for incorporation into a field-built apparatus. *Figure 38* is a cutaway view of the same type of washer and indicates the direction of air flow.

This washer consists of a rectangular steel chamber, closed at the top and sides and mounted on a shallow watertight tank of steel or concrete. Inlet baffles located at the air-entering end of the washer promote uniform air velocities thru the washer and minimize the spraying back of water into the entrance chamber as a result of air eddy currents. At the air-leaving end of the washer, eliminators are provided to remove entrained water droplets

Within the washer spray chamber two banks of opposing spray nozzles provide finely divided droplets of water uniformly distributed. After contacting the air, the water is collected in the tank and is returned to the sprays by a recirculating pump.

A central station washer may be designed for use as a

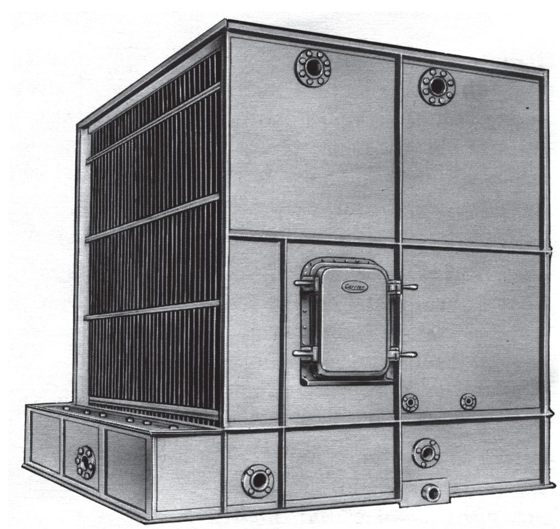


Fig. 37 — Central Station Washer

humidifier or as a dehumidifier. The physical arrangement is the same in either case. A dehumidifier is normally shorter in airway length than a humidifier.

Washers may also be obtained in a unitary design. A unitary spray washer, comparable in design and function to a central station washer, is shown in *Fig. 39.* Other types of unitary washers involve the wetting of a fibrous fill or set of pads located in the air stream.

The particular washer shown in *Fig. 39* operates at high spray chamber air velocities and is. therefore, smaller than a central station washer for a given air volume. *Figure 40* indicates the path of the air thru the unit components. The unit includes an inlet air mixing plenum, a vaneaxial fan, a diffuser section, a spray section and a rotating eliminator.

Two to six banks of sprays condition the air and clean it of dirt and other airborne particles. After contact with the air, the water drains from the spray section to a central tank from which it is recirculated.

APPLICATION

Air washers are primarily employed in industrial air conditioning applications. The use of sprays permits humidification, dehumidification or evaporative cooling, as required. In addition, sprays enable a degree of humidity control not possible with coils alone.

Washer equipment is effective in the removal of certain types of odors and dirt from the air. In applications where coils could become clogged with airborne solid particles, washers require a minimum of maintenance.

This flexibility of function is obtained at a relatively low installed cost of equipment per unit of air delivery. A large air capacity is realized from equipment of low weight.

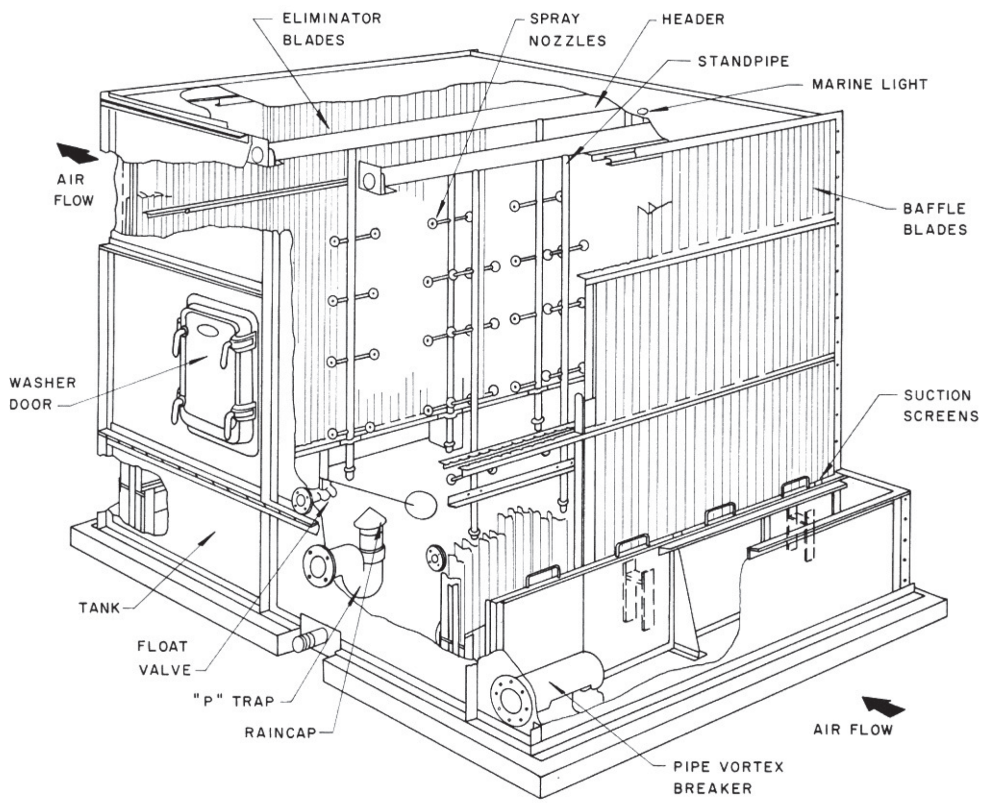


Fig. 38 — Central Station Washer (Sectional View)

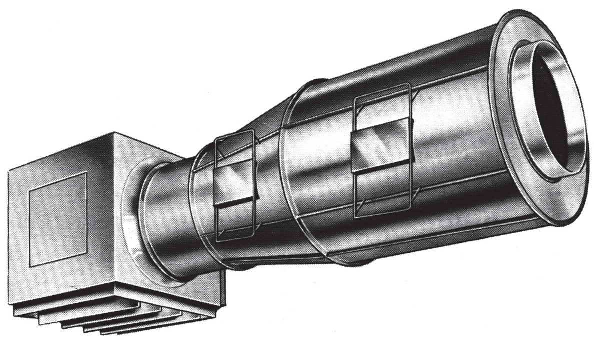


Fig. 39 – High Velocity Washer

This type of equipment is, however, open hydraulically and thus presents problems in piping design and system balancing. Refer to *Part 3* for a discussion of washer piping. Since the flow of air and water thin the apparatus is parallel and since a gravity return of water is usually employed in a dehumidifier application, pipe sizes tend to be larger in an open system and the piping system and insulation more expensive.

The spraying of water at high pressures such as are required in washer equipment produces a noise level high enough to be objectionable under some circumstances. Sound treatment is not usually required on equipment serving manufacturing areas or areas of high ambient sound levels. For more critical applications the need for sound absorption should be investigated

The unitary spray washer shown in *Fig. 39* requires considerably less space than a central station type and requires no special apparatus room. It is more flexible in meeting the necessities of plant layout change and is more adaptable to zoning. The salvage value is high and the operating weight low.

The central station washer installation results in lower fan noise levels and lower fan operating costs. Since central station washers are usually fewer in number and more centrally located than unitary washers, they require less piping when used as dehumidifiers for a given installation.

Central station washers may be obtained for air deliveries of 2000 to 336,000 cfm. Unitary spray washers are available in the delivery range of 7800 to 47,000 cfm.

Humidifier

A spray humidifier provides evaporative cooling thruout the year, as required, and heating during the winter season, if necessary. It is particularly suitable to applications where large quantities of sensible heat are to be removed, and where comparatively high relative humidities are to be uniformly maintained without the need for controlling dry-bulb temperature above a prescribed minimum. This type of washer equipment has been used extensively in the conditioning of industrial facilities engaged in the manufacture or processing of hygroscopic materials. Such industries include textiles, paper manufacturing, printing and tobacco processing.

A system of supplementary room atomizers is often used in conjunction with a spray humidifier in order to lower the first cost of the system. The psychrometrics of a combination system are outlined in *Par1.*

Spray humidifiers require the recirculation of water with no refrigeration. Recirculation occurs at the apparatus in the case of the central station washer. With the unitary washer, the recirculation of the water is produced centrally.

Dehumidifier

A spray dehumidifier provides sensible cooling and dehumidification during the summer season, evaporative cooling during the rest of the year, and heating, if

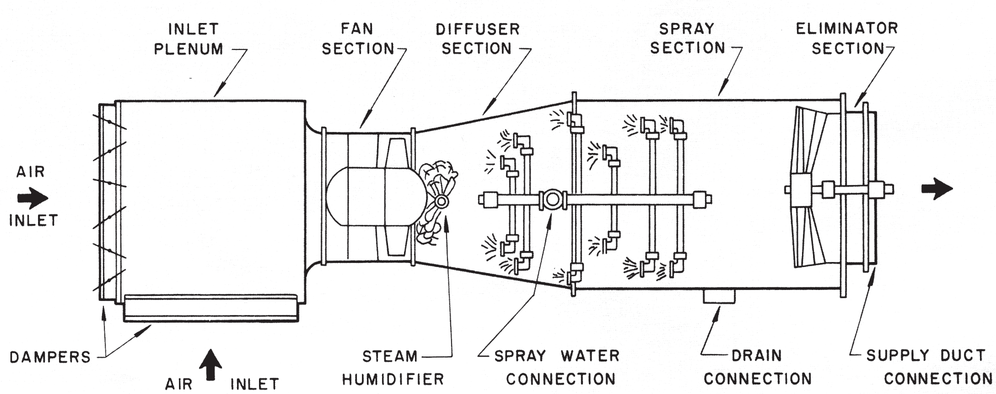


Fig. 40 – High Velocity Washer (Sectional View)

necessary, during the winter. It is used where lower relative humidities are to be uniformly maintained and where dry-bulb temperatures are to be controlled at a comfortable level. A source of chilled water is required for this application.

In a multiple central station system installation, the recirculated water quantity remains constant for each washer, and the chilled water is introduced in varying quantities at the suction of the recirculating pump during the dehumidifying season. See *Part 3.* The excess water returning to the washer tank is either pumped back to a central collection tank or, more commonly, drained from the washer to the central tank by gravity. If a gravity return is employed , a weir is required in the washer tank to maintain the water level in the tank. Rate of return in a pump-back application may be varied by a control valve actuated by the washer tank water level. A return pump should be sized to provide from l0 % to 20% more capacity than is required. In either case, the amount of chilled water admitted to the apparatus should be limited to a maximum of 90% of the recirculated water quantity.

Various central station tank arrangements are shown in *Fig. 41. Figures 41a and 41b* apply to gravity return dehumidifiers. *Figures 41c and 41d* are typical of pumped return dehumidifiers or evaporative cooling applications.

Although unitary spray washers may be arranged in the same manner as central station washers, they are usually supplied directly with chilled water with no recirculation at the unit. Spray density, therefore, varies with load. Water return is by gravity to a central collection tank.

During the months that refrigeration is not required, the chilled water pump serving the central station spray dehumidifier is idle.

Although efficient heat transfer is promoted by the direct contact of air and spray water in a washer, the parallel flow of air and water is less conducive to heat transfer than the counter flow process possible with a coil. *Figure 42* illustrates a method of obtaining a counter flow process with a two-stage spray dehumidifier. Flow is parallel thru each individual stage. Such an arrangement may permit a higher chilled water temperature or a smaller water flow.

UNIT SELECTION

The selection of a washer includes the determination of optimum washer size and dimensions and the establishment of the recirculated spray water quantity and pressure.

In the case of a dehumidifier, a study of the economic effects of a washer selection on other components such as piping or refrigeration equipment may be required. Increasing the recirculated water quantity or decreasing the washer face velocity by selecting a larger washer can permit operation at higher chilled water temperatures or at lower chilled water quantities.

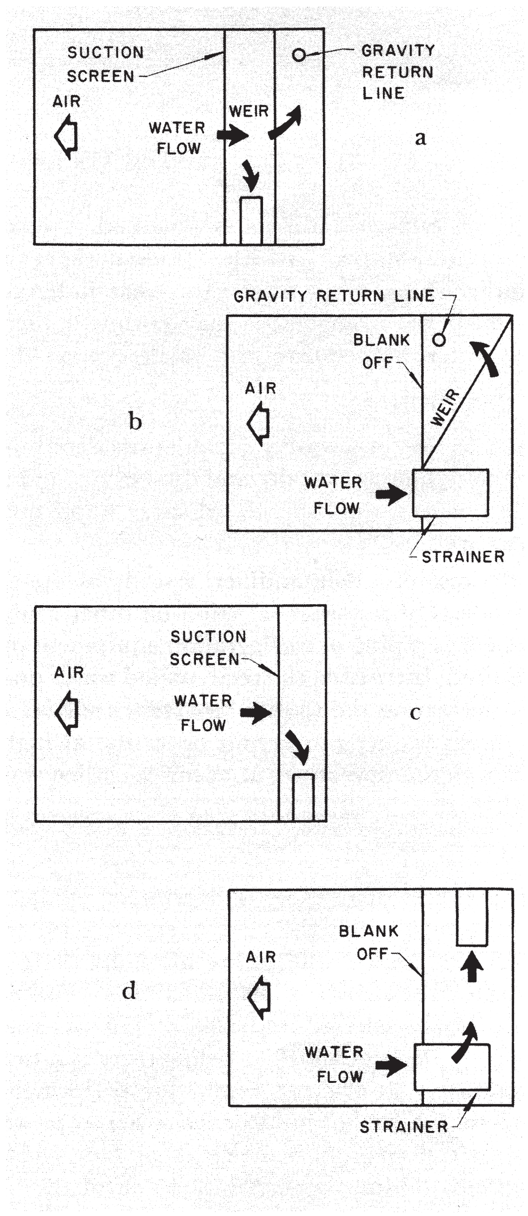


Fig. 41 — Washer Tank Arrangements (Plan View)

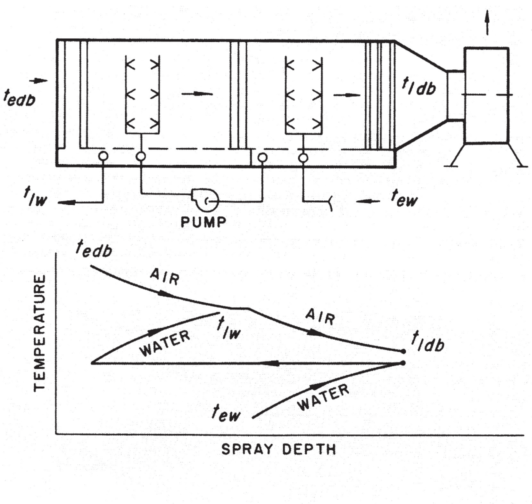


Fig. 42 — Two-Stage Counterflow Washer

Unit Size

The face area of a washer is determined by the design air quantity and the recommended maximum face velocity. Dehumidifiers are normally designed to operate at velocities of 300 to 650 fpm. Humidifiers are usually selected in the 300 to 750 fpm velocity range. Velocities above or below these limits are not conducive to efficient eliminator performance. Therefore, if a washer must be oversized to provide for future capacity and if the resulting face velocity is less than 300 fpm, a partial blank-off of the face area is suggested to increase the velocity until that time when full capacity is required. Similarly, if volume control is used to maintain space conditions ,the air velocity should not be allowed to drop below 300 fpm.

For maximum economy and flexibility of control, it is suggested that washers be selected at a face velocity as near as possible to the recommended maximum.

With an approximate face area determined, several washers of various heights and widths may be selected. First cost of the washer is usually minimized if it is selected as square as possible, with the height approximately equal to the width. However, at washer heights above a specified maximum the manufacturer may stack eliminators, thus in effect creating two washers. It is preferable economically to select the washer with a height below this maximum, even if the washer width then exceeds the height.

Washer saturation efficiency or contact factor decreases as face velocity increases. Thus, for a required air temperature rise, slightly more air is required at higher washer face velocities. However, the effect is not economically significant enough to justify lower washer face velocities.

The unitary spray washer *(Fig. 39)* operates at velocities up to 2600 fpm with efficient elimination of entrained moisture. This type of washer is rated to handle a nominal air quantity, and selections are made in tile range of 75% to 105% of nominal.

Spray Water

Washer saturation efficiency and contact factor are determined by various spray characteristics in addition to face velocity. These characteristics include the number of spray banks and the spray water pressure. At a given spray pressure, spray water quantity may be varied over a relatively wide range with little change in contact factor or saturation efficiency. This can be accomplished with different combinations of spray nozzle orifice size and number of nozzles.

Spray pressures usually lie in the range of 20 to 40 psig, with the higher pressures producing higher saturation efficiencies. Dehumidifiers normally require lower spray pressures than humidifiers.

At a given recirculated water quantity the fewer the number of spray banks, the greater the saturation efficiency since the spray pressure is greater. However, in the design and rating of central station washers, tile number of spray banks available is usually standardized and limited.

Optimum dehumidifier efficiency is usually obtained at a spray water quantity of approximately 5 gpm per square foot and a pressure of 25 psig. The spay density may vary from 3 to 11 gpm per square foot without an appreciable effect on the performance, providing the 25 psig nozzle pressure is maintained.

Humidifier spay densities vary from 2.25 to 3.0 gpm per square foot depending on the number and size of nozzles used.

Evaporative cooling applications require only a knowledge of size and saturation efficiency to complete the selection. However, for a dehumidifier selection the relation between leaving air wet bulb temperature and recirculated water temperature after air contact should be known. This information is necessary in order to calculate the quantity of chilled water required at a given temperature. *Chart 8* illustrates such a rating.

The unitary spray washer may be selected at various water quantities. A selection of spray banks is therefore available so that a range of contact factors may be obtained. Dehumidifier ratings are based on the apparatus dewpoint concept, as may be fan – coil unit ratings. A typical dehumidifying performance for a given unit size is shown in *Chart 9*.

Recirculating water pump heads for central station washers usually range from 50 to 85 ft wg, provided the pump is close to the washer. The pump head is primarily determined by spray nozzle pressure.

Fouling factors used for selection of refrigeration equipment used with washer equipment should be minimum of .001. See *Part 5*.

Atmospheric Connections

No correction to wash ratings is necessary for applications at altitudes above sea level. However, the design air quantity should be determined as described in *Part 1,* and the air side pressure drop of the washer adjusted as outlined in *Part 2*.

The fan selection should be in a accordance with the suggested procedure found in *Chapter 1* of this part. Motor selection at high altitudes is described in *Part 8*.

CHART 8—SPRAY DEHUMIDIFIER RATINGS (CENTRAL STATION)

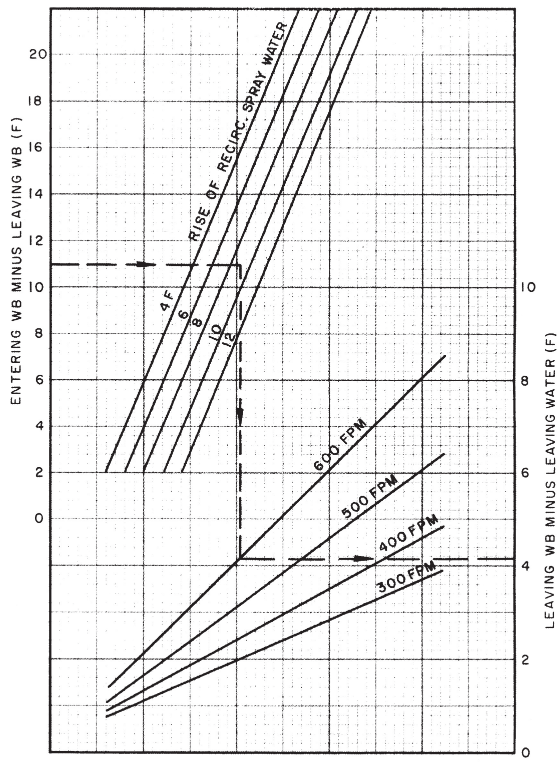
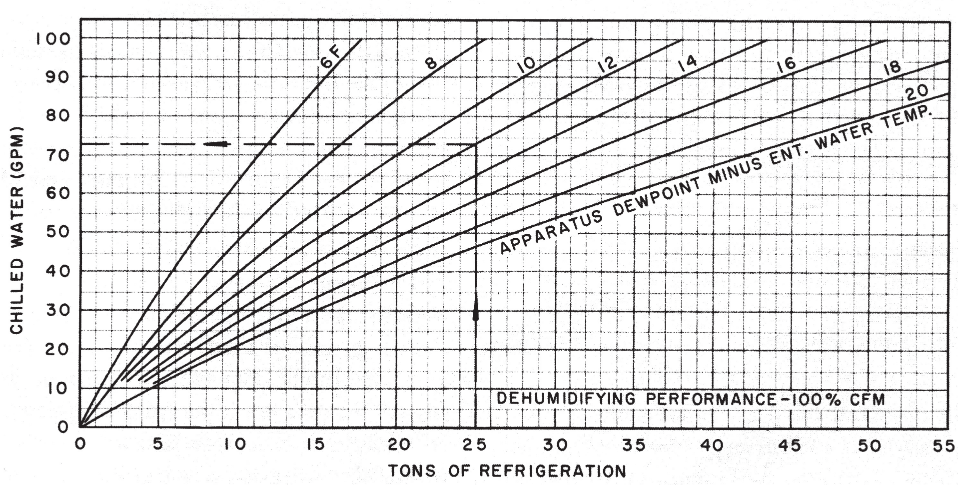


CHART 9—SPRAY DEHUMIDIFIER RATINGS (UNITARY TYPE)



ACCESSORIES

Flooding nozzles

For applications where solid airborne particles may accumulate on eliminator blades, flooding nozzles may be provided to continually flush the blades with recirculated water. Flooding nozzles may also serve the baffles at the entering face of a central station washer. Baffle sprays, however, are usually necessary only in applications with large quantities of air born list, such as textile mills.

Eliminator flooding nozzles usually operate at spray pressures of 3 to 10 psig for central station washers and 5 to 20 psig for unitary washers. With the central station type of washer a spray water quantity of 4 gpm per row per foot of washer width is suggested. One row is generally required for each eliminator section. The flooding of eliminator blades should be limited to those blades of at least six bends.

Baffle spray nozzles may be designed for spray pressures of 5 to 15 psig, and should be spaced to provide effective baffle coverage at a spaced to provide effective baffle coverage at a spray water quantity of 3 to 6 gpm per foot of washer width per pipe header. Headers should be spaced 2 to 3 feet apart at the entering face.

Flooding nozzle water requirements may be furnished by a separate recirculating pump or may be delivered by the main recirculating pump. If the latter means is chosen, the flooding nozzle water quantity should be added to that of the main sprays and the total then used to select the pump.

Water Cleaning devices

In order to insure proper spray nozzle operation and a minimum of manual cleaning and maintenance, foreign matter from the air stream and from eliminators and baffles should be removed from the spray water.

Two types of cleaning devices are commonly employed for this purpose: stationary screens and automatic self-cleaning strainers. Self-cleaning strainers are usually the rotating drum or endless belt type.

Stationary screens are located in the washer tank so that spray water must pass thru them before being recirculated. Cleaning the screens is a manual operation and can be facilitated by using two screens in series, supported by independent screen guides. The screen openings should be smaller than the spray nozzle orifice size. The washer tanks shown in *Fig. 41a* and *41c* should be equipped with stationary screens.

An endless belt self-cleaning strainer may be used with or in place of the stationary screens and is suitable mainly for applications where foreign matter particles are of a relatively large size. It operates continuously, collecting the particles on a belt and then flushing them

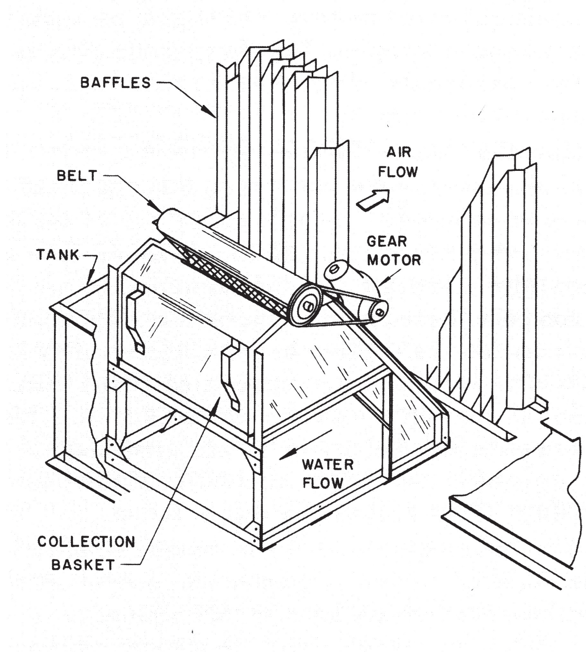


Fig. 43 — Belt Type Water Strainer

with recirculated or city water from the belt into a basket. If recirculated water is used, the requirement should be added to that of the main spray and flooding nozzles in order to determine the required pump capacity. A blet strainer can be located within the washer tank (*Fig. 41b* and *41d*). A belt strainer is shown in *Fig 43*.

The rotating drum strainer is installed in a central storage and collection tank. It is a more efficient cleaning device than the stationary screen or belt type strainers. For this reason it is particularly suited for use with a unitary spray washer system where all water is returned to a central location and where the tubes of a water cooler and various control valves must be protected from foreign particle accumulations. *Figure 44* illustrates a rotating drum strainer.

With this method of cleaning water is filtered continuously thru a perforated drum. Accumulations of residue on the drum surface cause the water level to ruse in order to seek more perforations. The variations in water level control the periodic drum and to remove the waste matter to a collecting basket.

Spray Water Heaters

Spray water heaters may be required in winter when the mixing of outdoor and return air upstream of the washer cannot be controlled to produce the washer entering wet-bulb temperature required to maintain room design conditions with an evaporative cooling process. This condition may occur on very cold days or where minimum outdoor air requirements are relatively high,

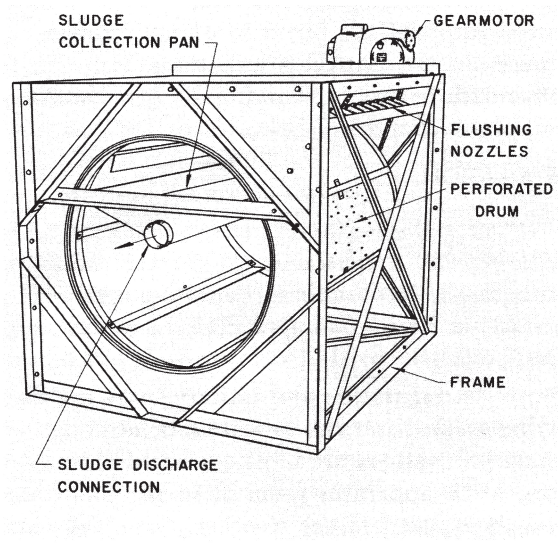


Fig. 44 — Rotating Drum Water Strainer (Water Entering Side)

particularly with high room relative humidities and/or high room sensible heat ratios.

After a shutdown, as over a winter weekend, it may take some time to bring the room humidity up to design conditions when operating on an evaporative cooling cycles, even when the outdoor air quantity is reduced to damper leakage. Therefore, the spray water heater is used to add moisture to the air at approximately the same dry-bulb temperature. The heater thus provides a deviation from an adiabatic saturation process (*Fig. 45*).

Both steam ejector heaters and closed water heaters are available for heating spray water in central station washers. The steam ejector heater is a perforated steel pipe, closed at one end and submerged in the washer tank. Low pressure steam is admitted directly to the washer tank water at a controlled rate. The closed water heater is located on the discharge side of the recirculating pump and is installed in parallel with the main recirculating supply line. It is selected to heat a minimum quantity of spray water and requires suitable service and balancing valves. The closed water heater produces less noise than the steam ejector heater and enables recovery of steam condensate. However, the closed water heater is more costly to purchase and install.

A spray water heater may be sized on the basis of requirements calculated at the particular conditions encountered. Its capacity may also be calculated by determining the steam quantity required to heat and humidity the minimum outdoor air, or outdoor air damper leakage, from outdoor to room design conditions. The latter method provides sufficient capacity to maintain room design conditions during the period following equipment start-up.

The high velocity unitary washer utilizes a steam grid humidifier for humidity control under the conditions described above. Since steam is released directly to the air, relatively little sensible heating is accomplished.

Weirs

A weir is employed in a central station spray dehumidifier tank to insure a minimum submergence of the recirculating pump suction pipe and to maintain a water seal under the eliminators. During the evaporative cooling season there is no flow over the weir. During the dehumidifying season, however, the flow is equal to the chilled water admitted to the recirculating system plus the moisture removed from the conditioned air.

Weir tanks may usually be obtained from the washer manufacturer. If a concrete tank is to be used, the length of the weir may be calculated from the Francis formula for sharp-crested rectangular weirs,

$$Q = 3.33 \times L \times H^{1.5}$$

The flow Q is expressed in cubic feet per second. The length L and the head H are in feet. If the weir is to have end contractions, the length should be reduced by 0.1 x H for each end contraction, for formula use.

The sharp-crested concrete weir may be obtained by bolting a steel angle to the flat crest.

A flow rate of 5 gpm per foot of weir length is common for dehumidifier tanks.

Isolation

A central station washer requires no vibration isolation. The supply air fan isolation requirements should be investigated, however, with regard to the ambient sound levels thruout the building. Unitary washers seldom require vibration isolation for industrial applications but a vibration analysis may be necessary for critical installations. Isolation recommendations may be found under *Fan-Coil Equipment* in this chapter.

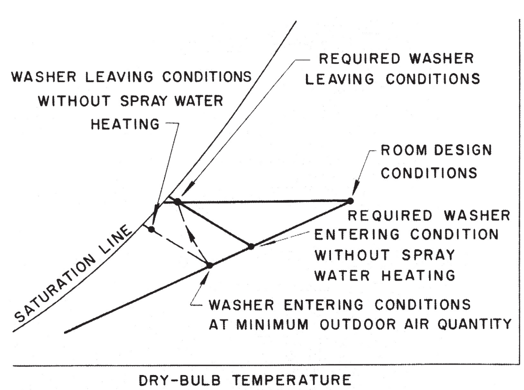


Fig. 45 — Effect of Spray Water Heater

INSTALLATION

Location

The economic and sound level considerations pertaining to the location of air handling apparatus, as discussed in *Part 2,* are applicable to washer equipment.

Both central station and unitary spray apparatus may be located indoors or outdoors, although central station washers are most commonly located indoors, in an apparatus room or in the conditioned space. Ii exposed to the weather, a central station washer should operate with no water level maintained in the tank, and the fan motor, drive and bearing should be suitably chosen and protected.

Central station equipment is floor-mounted while the unitary washer may be either floor-mounted or suspended from above *(Fig. 46).*

As with fan-coil equipment the availability of outdoor air and the ease of air return to the apparatus is of primary concern in selecting a washer location. Outdoor air intakes should, if possible, be located and oriented so that they do not face nearby residential areas or walls of spaces where noise would be objectionable. Air may be returned thru a duct system but, if it is returned directly to the apparatus, the apparatus should be located so as to receive return air from the area it serves.

Another important location consideration is the availability of space. Particularly in industrial production areas space may be difficult to acquire. Limited head room and interferences such as electrical equipment, conveyors or belt drives may also present problems.

In addition, the location and orientation of the washer should be guided by the following considerations:

1. A spray dehumidifiers should be located so that the gravity return of water to the central tank is possible. If this condition cannot be met, a pumped return should be employed.
2. Adequate building openings and passages should be available for the admittance of large equipment. If this consideration is overlooked, special openings in the building may later be required.
3. Locating a spray dehumidifier below the refrigeration equipment or pumping return water to a lower elevation may lead to problems of siphoning or overflow at shutdown. If such an arrangement is necessary, consideration should be given to the checking of water back-flow tendencies and the breaking of a siphon.
4. The ability of a roof, floor or combination of structural members to withstand the operating weight of a washer should be investigated.
5. A washer should be located and oriented so that the simplest possible duct layout results.
6. Appearance should be considered. For example, a washer mounted on a flat roof may be less noticeable if located some distance from the building perimeter.

Layout

Figure 46 illustrates several layout alternatives for a high velocity unitary washer. A typical central station washer apparatus room layout resembles that shown for coil equipment in *Part 2.*

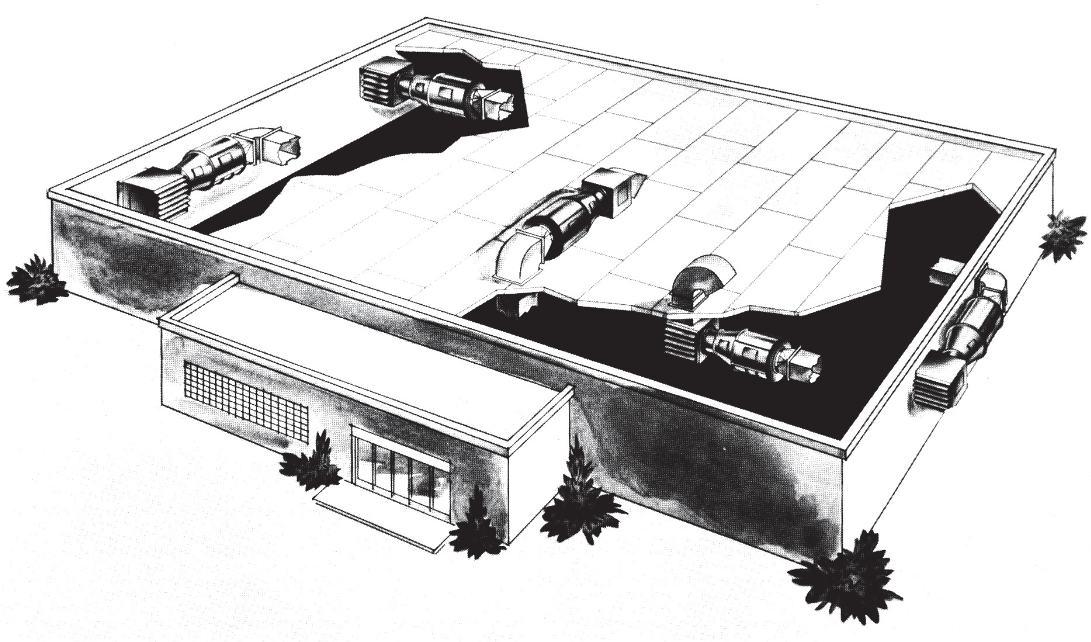


Fig. 46 — Unitary Washer Installation

Central station washers should be provided with inlet plenums of adequate airway depth to promote outdoor and return air mixing and to minimize air eddy currents at the inlet face of the washer. The plenum chamber on the leaving air side of the washer should be large enough to provide unrestricted air flow to the fan at uniform eliminator velocities. Plenums downstream of the washer should also permit easy cleaning after removal of the eliminator blades for both central station and unitary apparatus.

Sufficient space should be provided around the washer for maintenance access, particularly on the side where access doors and piping connections are located. Minimum clearance should be provided on the far side for cleaning, painting and for the application of insulation if required. If suspended well above the floor, unitary washers may require catwalks.

Access doors should be installed between central station apparatus components as required for proper maintenance and service.

A mounting base at least two inches high should he provided for central station equipment. A base provide a level and uniform bearing surface for the tank, prevents damage to the tank or to the insulation under the tank from water seepage, and increases the tank water level available for priming the recirculating pump.

If a concrete tank is designed for a central station washer, it should be of reinforced construction and provided with pipe sleeves, baffle and eliminator supports and anchor bolts, as required. In addition the plenum at the leaving air side of the washer should be provided with a curb at least four inches high.

The recirculating water pump may be located in the air stream entering the washer or outside of the washer casing.

Marine lights should be provide within the washer and between components of a central station washer.

In addition to the washer piping details shown in *Part 3,* the following suggestions apply:

1. Floor drains should be provided in the entering and leaving air plenums, near the recirculating pump, near the outdoor air intake and as required for the cleaning of filters or other components. Usually the drain in the leaving air plenum requires a deep seal trap.
2. If a pumped return is used for a spray dehumidifier, a continuously running 1/2 inch bleed line from the return pump discharge to the tank prevents overheating of the water in the pump when the return control valve is closed.
3. Because the water in a washer tank is relatively shallow, a well designed vortex breaker on the pump suction pipe is required.

Insulation

The top and sides of a spray dehumidifier should be insulated as required to prevent condensation on the apparatus and to minimize heat transfer. A central station humidifier should be similarly insulated if the dewpoint for the return air is higher than the spray water temperature, such as occurs when supplemental atomizer systems are employed.

The high velocity unitary washer described previously should be completely insulated, regardless of the application.

Vapor harriers must be applied to the high dewpoint side of the insulation to prevent condensation forming on the metal surface of the unit.

A thickness of cork insulation may be required beneath the washer tank. If used, the cork layer should be coated on both sides with sealing compound and positioned on the tank pad before the unit is installed.

Washers located outdoors should be insulated, vapor sealed and weatherproofed. The insulation on the top surfaces should be slightly crowned so that water will run off.

Water and steam riser insulation in industrial applications is sometimes subject to damage from trucks and material handling equipment. If such is the case, a sheet metal shield around the insulation is suggested, from the floor to a height of several feet.

CONTROL

The function of controls is to produce a balance between the air conditioning load and the apparatus capacity in order to maintain room design conditions.

Apparatus control may be accomplished in one or a combination of two ways:

1. Varying the supply air volume at a given air condition.
2. Varying the air condition with no change in volume.

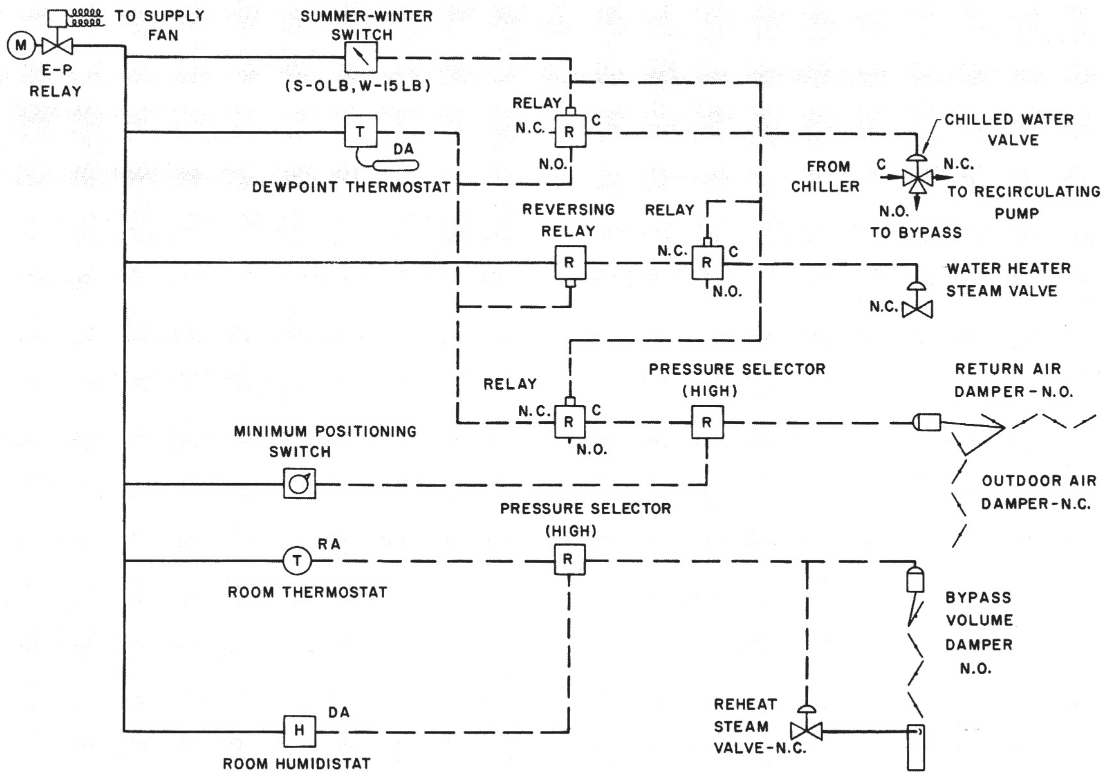


Fig. 47 — Central Station Dehumidifier Control

The condition of the air may be altered by such methods as spray water temperature variation, air reheating, spray water heating, washer bypass, spray throttling, variation of the outdoor and return air mixture proportions, and air humidification, as with a steam grid humidifier or an atomizer system.

A central station air washer operating on a dehumidifying cycle normally utilizes spray water temperature variation, air volume reduction and air reheating. See *Fig.* 47 for the simplified control diagram. A dry-bulb thermostat, located on the leaving air side of the washer and set to maintain the leaving dry-bulb condition necessary to achieve the required leaving air dewpoint, controls the chilled water valve supplying the recirculating water system. The spray water temperature is thus controlled.

Dewpoint control is practical because the difference between dewpoint and leaving air dry-bulb temperature is small as a result of the high contact factor of the dehumidifier.

Air volume and reheat control is obtained by a thermostat and humidistat together controlling a reheat coil bypass volume damper and a reheater steam valve in sequence. Closing the volume damper reduces the supply air volume to a predetermined fraction of full load air delivery, usually from 60% to 85%, depending on the fan characteristic and the allowable maximum air pressure drop thru the heating coil. A further reduction in room load causes the reheater valve to begin opening.

*Figure 48*a illustrates a typical spray dehumidifying process at full load. *Figure 48b* shows the temperature relations at half load. The entering spray water temperature has been increased to maintain a relatively constant apparatus dewpoint.

Chilled water supply to the central station dehumidifier recirculating system can be controlled by a two-way throttling valve or a three-way diverting valve. Use of a two-way valve on a multiple washer system may necessitate a pressure bypass line anti valve at the central collection tank in order to minimize line pressure fluctuations.

Dewpoint control of a central station air washer operating on an evaporative cooling cycle is achieved by controlling the outdoor and return air mixture condition and by operating the spray water heater, if necessary. Zone control may be identical with that control employed when on the dehumidifying cycle.

A measure of humidity control may be obtained during a period of refrigeration shutdown by cycling the recirculating pump and/or the baffle or eliminator spray pump, if any.

In a high velocity unitary washer application, room

conditions are controlled directly by a combination of spray throttling, air reheating and, if necessary, humidification. *Figure 4*9 is a control diagram for units operating with dehumidification control for an all air system. During the dehumidifying season the outdoor air dampers are in a minimum position; the reheat valve is con trolled by the room thermostat alone; the spray throttling valve is controlled thin the pressure selector, and the humidifier is controlled by the humidistat, providing the thermostat is satisfied. When operating on the evaporative cooling cycle, the outdoor and return air dampers are controlled by the room thermostat: the reheat valve is controlled thru the pressure selector; the spray throttling is controlled by the humidistat, and the humidifier is controlled as in the dehurnidifying cycle.

Since spray throttling is always utilized with this system, a pressure bypass line and valve are usually required at the central water collection tank to minimize fluctuation in line water pressures.

General Control Considerations

For industrial applications, particularly in the case of process air conditioning, room controls are often mounted within a cabinet provided with a small fan. The circulation of room air thru the cabinet provides a constant and positive sampling by the control sensing devices.

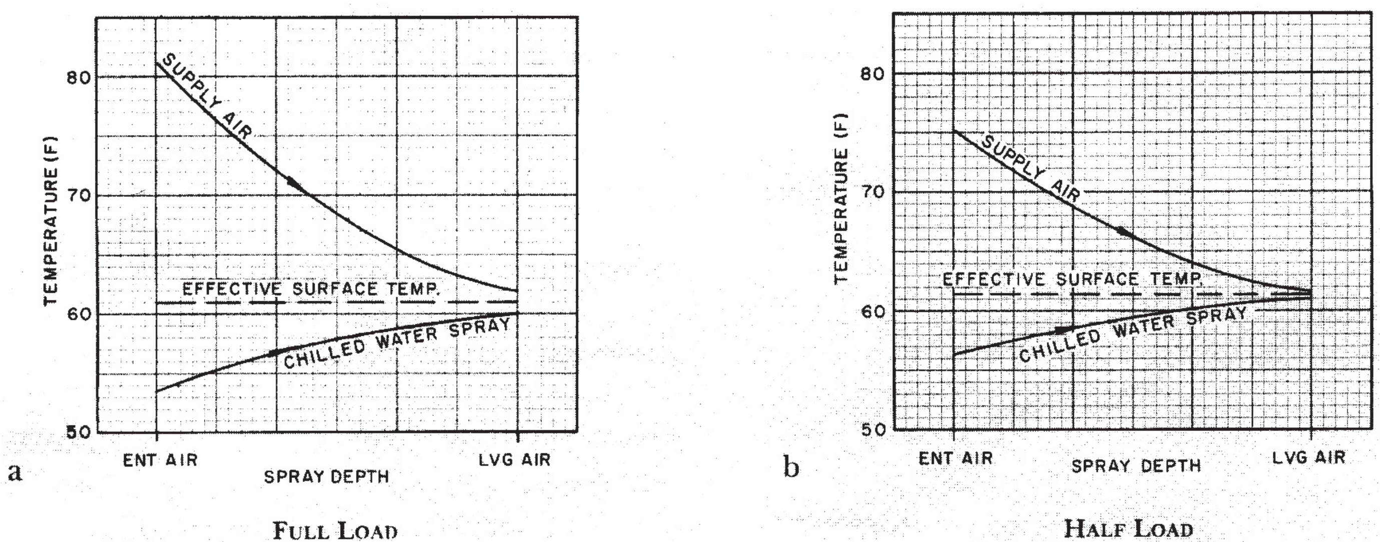


FIG. 48 – TYPICAL SPRAY DEHUMIDIFIER PROCESSES

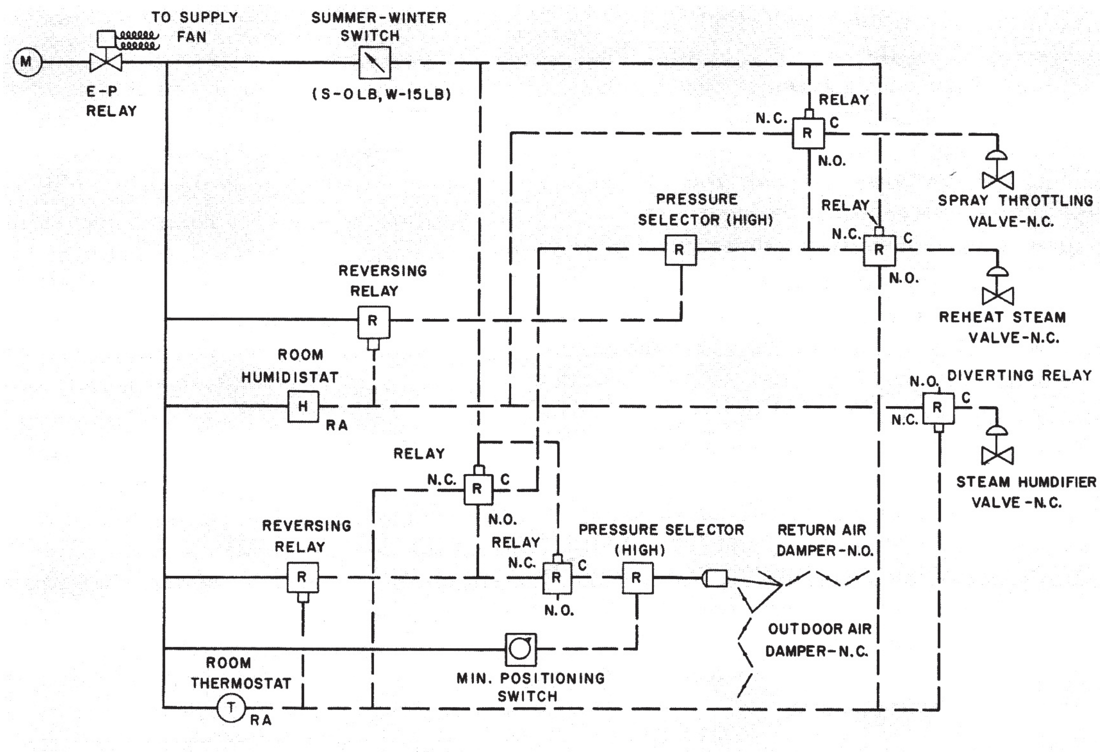


Fig. 49 — Unitary Washer Control (Dehumidifier All Air)

If large open spaces are to be conditioned, the area served by each set of room controls should be limited in order to maintain an adequate control response. Maximum areas of 10,000 square feet for a temperature zone and 8000 square feet for a humidity zone are suggested.

Control accuracy and response are for also affected by the air circulation in the conditioned space. A maximum of ten minutes for a complete change of air is suggested, and four to eight minutes is preferred.

CHAPTER 3. UNITARY EQUIPMENT

A unitary air conditioning unit, sometimes referred to as packaged equipment, consists of one or more factory-fabricated assemblies designed to provide the functions of air moving, air cleaning, cooling and dehumidification. The functions of heating and humidifying are also usually possible with such equipment. Heat pump versions are available for most types of apparatus.

Unitary equipment includes a direct expansion or chilled water cooling coil and a compressor-Condenser combination or water chiller in addition to fans, auxiliaries and internal wiring and piping. If more than one assembly is required, the separate assemblies are designed for use with each other, and combined equipment ratings are based on matched assemblies of equal or differing nominal capacities.

The design of unitary equipment is often styled for installation within the conditioned space.

It is the purpose of this chapter to guide the engineer in the practical application and selection of unitary equipment.

TYPES OF EQUIPMENT

Unitary equipment may be classified as either a self-contained or a split system. A self-contained unit houses all components in a single assembly. Split system equipment incorporates the following assemblies

1. A coil and compressor combined with a remote condenser.
2. A coil combined with a remote condensing unit.
3. A coil combined with a remote water chiller.

A self-contained unit is illustrated in *Fig. 50.* The self-contained concept is further described in *Fig. 51. Figure 52* shows an air-cooled condensing unit — one component of a two-component split system as described in Item 2 above.

The use of matched components differentiates unitary equipment from the fan-coil equipment discussed in *Chapter 2* of this part. Unitary equipment thus affords less flexibility of arrangement and less choice of cooling

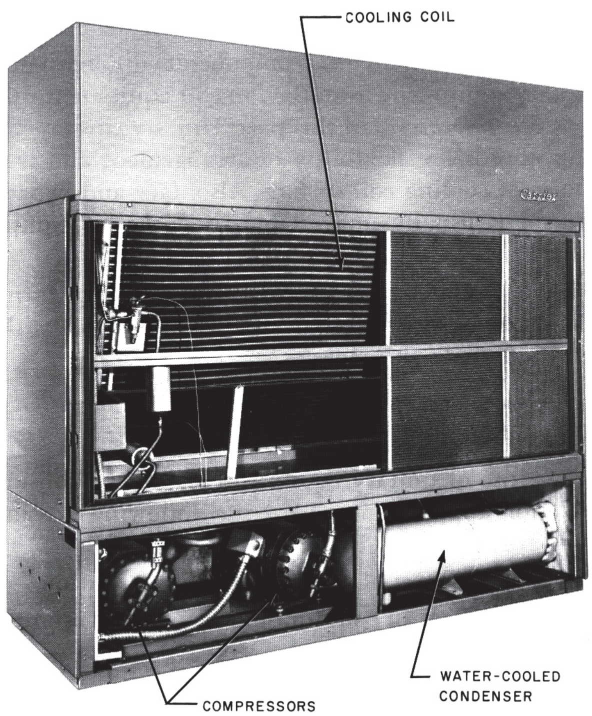


FIG. 50 — SELF-CONTAINED UNIT

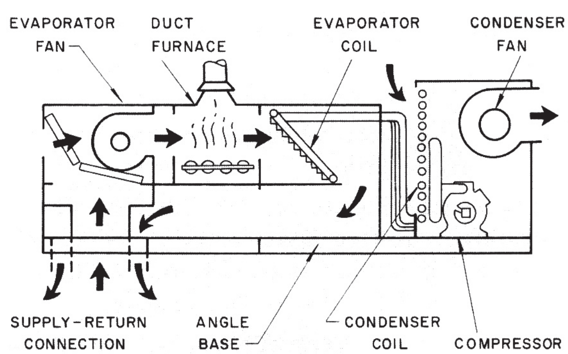


FIG. 51 — SELF-CONTAINED UNIT

Fig. 52 — Air-Cooled Condensing Unit

coil surface. Also, face and bypass control is usually unavailable in packaged equipment.

Split system apparatus provides in packaged form a measure of flexibility not usually obtainable with self-contained equipment.

APPLICATION

The use of unitary equipment should be considered for applications where the following advantages are of primary importance:

1. Low first cost of equipment and installation.
2. Immediate air conditioning benefits and prompt delivery.
3. Ease of installation or removal, if necessary, with a minimum of disturbance.
4. The ability to provide air conditioning in increments without cost penalty.
5. Economical operation during periods of nonuniform loading.
6. High salvage value and longer warrantee periods.
7. Simplified field engineering.
8. Factory assembly of balanced and tested components.

Packaged equipment is particularly well suited to applications requiring summer cooling only, and is readily used in conjunction with existing or separate heating facilities of sufficient capacity.

Such equipment may effectively augment central station apparatus by serving relatively small areas with special design requirements. Typical applications of this nature are laboratories and dining areas.

Applications completely conditioned with unitary equipment include existing office buildings and hotels, motels, shopping center tenant areas, department stores, industrial facilities and residences.

Equipment components are usually matched to provide 300 to 500 cfm per ton of air conditioning at sensible heat ratios of 0.65 to 0.85, in the case of self-contained equipment. Therefore, packaged equipment is most economically applied where these values are specifically required. As mentioned previously some additional application flexibility may be obtained by employing split system equipment. Sensible heat ratios as high as 0.95 are attainable. Such equipment also affords greater choice of location and mounting method.

Self-contained equipment is commonly available in capacities up to 60 tons, while up to 75 tons may be obtained with a split system. The trend has recently been toward larger packaged equipment .

Water-cooled, air-cooled or evaporative condensing may be utilized with unitary apparatus.

STANDARDS AND CODES

Applicable provisions of the American Standard Safety Code B9.1, ARI Standard 210 and Underwriters' Laboratories Standards govern the testing, rating and construction of unitary air conditioning equipment.

The application and installation of such equipment should conform to pertinent government agency regulations and to all codes and laws prevailing at the job site.

UNIT SELECTION

SELECTION RATINGS

Unit size is usually determined by the required cooling capacity and air quantity, adjusted to suit the sensible heat ratio. Cooling ratings present total and sensible heat capacities, based on air quantity, evaporator entering air wet-bulb temperature and, in the case of water-cooled equipment, condensing temperature. A typical cooling rating table is illustrated in *Fig. 53.* Although tabular cooling ratings are most common, some manufacturers present graphical data in place of, or in addition to, tabular ratings.

Cooling ratings *(Fig. 53)* may be expanded to apply to more than one evaporator entering air dry-bulb temperature. If they are not expanded, deviation corrections are usually suggested. Cooling rating also may indicate grand sensible heat factor rather than total sensible heat capacity.

For air-cooled condensing or evaporative condensing, selection ratings are normally based upon condenser entering air dry-bulb temperature or wet-bulb temperature respectively, instead of condensing temperature.

Self-contained equipment is rated as a system with no individual component ratings. A split system apparatus is usually rated both as an individual item of equipment and in combination with its intended components. For example, an air-cooled condensing unit may be rated in terms of cooling capacity available from the package when the condensing unit is used with a particular fan-coil unit at different evaporator wet-bulb and outdoor dry-bulb temperatures.

EVAPORATOR AIR		CONDENSING TEMPERATURE (F)											
		90			100			105			110		
Qty (cfm)	Ent Wet-bulb (F)	Total Cap. (1000 Btuh)	Sens Heat Cap.* (1000 Btuh)	Compr Motor Power Input (kw)	Total Cap. (1000 Btuh)	Sens Heat Cap.* (1000 Btuh)	Compr Motor Power Input (kw)	Total Cap. (1000 Btuh)	Sens Heat Cap.* (1000 Btuh)	Compr Motor Power Input (kw)	Total Cap. (1000 Btuh)	Sens Heat Cap.* (1000 Btuh)	Compr Motor Power Input (kw)
4500	72	204	97	12.8	198	95	14.0	194	94	14.6	190	92	15.2
	67	186	119	12.4	181	117	13.7	178	116	14.2	173	114	14.9
	62	169	142	12.2	165	139	13.4	162	138	13.8	158	136	14.4
6000	72	213	108	12.9	207	106	14.2	203	105	14.8	198	103	15.4
	67	196	136	12.7	190	134	13.9	186	133	14.4	182	131	15.1
	62	179	164	12.4	173	162	13.6	170	161	14.0	167	159	14.6
7500	72	220	118	13.0	214	115	14.3	210	114	14.9	204	113	15.6
	67	204	151	12.7	197	150	14.0	192	148	14.6	188	146	15.2
	62	185	184	12.4	180	180	13.7	176	176	14.2	173	173	14.8

*Sensible heat capacity is based on 80 F entering air dry-bulb temperature.

Fig. 53 — Typical Ratings (Water-Cooled Self-Contained Units)

ECONOMICS

The cooling ratings described above are based upon the capacities of components in balance with each other. It is therefore usually unnecessary to determine component balance capacities when selecting packagedequipment. However, if equipment is to be economically selected for use with unmatched components, the determination of the component balance points and the subsequent selection analysis should be pursued as described in *Part* 7.

The economical balance of component capacities may be upset if the grand sensible heat factor required for an application differs considerably from that characteristic of the package. For example, with relatively high sensible heat ratios the desired air quantity per ton of capacity is also high, relative to that available from standard packaged equipment. With self-contained equipment, therefore, it may be necessary to provide oversized refrigeration components, in order to acquire the air appropriate air delivery. In this case an economical balance can be restored by designing for lower relative humidities, thus permitting a greater air temperature rise and a smaller air quantity. An alternate solution, applicable within limits, is to vary the evaporator air quantity. Split system equipment may be mixed as to nominal component capacities to produce an economical balance.

Economy of equipment selection may also be promoted by the following methods:

1. Select equipment to be fully loaded, taking advantage of room temperature swing, storage effects and reduced safety factors.
2. Avoid the arrangement of unit by zones where selections must be made for peak loads. Diversity benefits may be realized if more than one exposure is served by a unit.
3. Consider operation at relatively high condensing temperatures with possible savings as explained in *Part* 7.
4. Introduce the least outdoor air possible at peak apparatus load.

ATMOSPHERIC CORRECTIONS

Unitary equipment ratings are based on air at standard atmospheric conditions of 70 F and 29.92 in. Hg barometric pressure. For applications deviating significantly from this standard such as at altitudes exceeding 2500 feet, ratings should be adjusted for the difference in air density. The several corrections involved have been described elsewhere and may be summarized as follows:

1. Load calculations should be modified as described in *Part 1.*
2. Air side pressure drop should be adjusted in proportion to the air density ratio as outlined in *Part 2.*
3. If the apparatus includes an evaporator, the unit capacity should be corrected by entering the rating tables at a supply air quantity equivalent to standard atmospheric conditions. This procedure is similar to that described for fan-coil units in *Chapter 2* of this part.
4. Fan speed and brake horsepower should be adjusted as detailed in *Chapter 1* of this part.

The decrease in performance of air-cooled condensers at high altitudes and/or air temperatures, although in itself significant, produces only a slight deviation in the rating of combined components. This deviation may amount to from one to three percent at an altitude of 5000 feet.

COIL FREEZE-UP PROTECTION

The freezing of hot water coils located downstream of the cooling coil may occur during the cooling season. This is particularly possible in packaged fan-coil equipment because of the proximity of the heating and cooling coils and because of the relatively low settings usually employed on the low pressure compressor cutoff switch to prevent excessive cycling. At lower evaporator wet-bulb temperatures, equipment components balance at diminished suction temperatures. Hot water coil freeze-up may occur under these conditions.

Although coil freeze-up may be prevented by the draining of the hot water coil or by the use of a properly inhibited antifreeze solution as described in *Chapter 2* of this part, the use of a protective thermostat to cycle the compressor is suggested instead in the interest of economy. The thermostat should be mounted outside of the air stream with the bulb at the entering face of the hot water coil. An air temperature of approximately 35 F is suggested as a compressor shutoff point.

Coil freeze-up may sometimes be traced to reduced air quantities resulting from dirty filters in the apparatus.

INSTALLATION

LOCATION

Unitary air conditioning equipment may usually be located either outdoors or indoors. Possible specific locations include basements, crawl spaces, attics, garages, roofs, and on the ground as well as within the conditioned space or in an equipment room. Such equipment may be mounted on the floor, suspended from the ceiling or installed in a wall opening, transom or window. Since packaged apparatus is sometimes designed for a specific location such as on a roof or under a window, the manufacturer's literature should be studied for location recommendations. The suggestions in *Part 2, Part 7,* and *Chapter 2* of this part regarding equipment location are applicable to packaged equipment.

Although weatherproofing kits are available for most self-contained equipment, an indoor location is preferred. However, self-contained equipment designed specifically for outdoor use is available. With any equipment featuring Outdoor compressors, crankcase heaters should be employed to prevent the migration of refrigerant to the compressor and the damage which may result. For outdoor or indoor locations insulation considerations apply as noted in *Chapter 2* of this part.

LAYOUT

The references noted above should be consulted for suggestions dealing with equipment layout.

Unitary apparatus is available in both horizontal and vertical arrangements and it is usually designed for use with or without duct systems. However, distribution systems should be simple in design and limited in extent.

Often, unitary equipment may conveniently use the same air distribution system as an existing heating system. This is particularly true in residential applications. In such an installation appropriate shutoff or diverting dampers may be required if the heating and cooling functions are in parallel. Existing duct sizes should be checked for adequacy in handling the dehumidified air quantity.

For roof-top installations the roof must be of adequate strength, and the equipment weight should be evenly distributed on the support members. If any doubt exists as to the adequacy of support, a structural engineer should be consulted. Appropriate framing around roof openings, flashing, counter flashing and pitch pockets should be provided.

External vibration isolation of packaged equipment is seldom required because the individual Components are usually isolated within the cabinet.

However, for critical installations and light building construction, vibration isolation should be considered for unitary equipment as for any other type of equipment. Vibration isolation is discussed in *Chapter* 2 of this part. Isolation recommendations may also be solicited from the manufacturers of vibration equipment.

Layout and location of unitary equipment are influenced by the availability of service facilities such as gas, city water and electrical power.

CONTROL

The reduction of packaged equipment capacity at partial loads is usually effected by cycling the compressor or compressors in accordance with the setting of a room dry-bulb thermostat. decreasing compressor capacity by the unloading of cylinders is another widely employed method of control.

Unit fans may be created continuously or cycled with the compressor. Continuous operation of fans provides continuous air circulation. However, alternately condensing moisture from the air and reevaporating coil moisture produces fluctuations in room humidity conditions. Cycling of fans requires the use of a room thermostat rather than a return air thermostat.

The desirability of controlling equipment capacity and the unavailability of face and bypass control may affect adversely the equipment capacity for latent heat removal. For example, with a single compressor serving a single evaporator coil, cylinder Unloading causes an increase in the grand sensible heat ratio and thus a relative decrease in the latent heat capacity. This occurs at a time when the

opposite effect is usually desired.

This effect may be overcome by the use of multiple compressors with multiple coils or coil circuits operating on signals from a two-step thermostat. This permits an improvement in latent heat removal at partial loads. Coils may be of equal or unequal capacities. in either case, some additional latent capacity is obtained by the decrease in air quantity over the operative coil as the inoperative coil dries. With unequal coils, where the larger is the first to be removed from operation, additional latent capacity is also obtained thru the lower sensible heat ratio of the smaller coil and the prolonged ''on'' period of the operating cycle.

Multiple compressors are usually obtainable only on equipment of ten tons capacity, or greater.

Loss of latent capacity at partial loads and the resulting fluctuations in room conditions are intensified by the oversizing of equipment. Fully loaded equipment provides the best insurance of maintaining reasonable humidity conditions at partial loads. Unit air deliveries may also be varied from nominal to obtain the most desirable full load sensible heat ratio, and therefore the best possible latent capacity at partial load.

Under special conditions where accurate control of temperature and humidity is required, packaged equipment may be easily adapted for control by reheat or humidification.

The necessity and means of condensing pressure control is discussed for various condensing methods in *Part 7.*

CHAPTER 4. ACCESSORY EQUIPMENT

This chapter presents practical information to guide the engineer in the application and layout of air cleaning and heating devices, as used in conjunction with air conditioning systems.

AIR CLEANERS

The control of air purity consists of reducing or eliminating unwanted particulate or gaseous matter from the air supplied to a space. This is a function of the air conditioning system. However, normal applications are concerned with particulate matter only.

Effectively applied air cleaners can materially reduce operating expenses and increase productivity. Specific benefits include:

1. The reduction of building cleaning costs — an item otherwise accounting for as much as forty percent of total operating expenses.
2. The reduction of employee absenteeism — a result of the removal of bacteria, viruses and allergens from the air.
3. An increase in employee efficiency.
4. An increase in product quality.
5. An increase in the life of machinery or equipment.

CONTAMINANTS

Air is contaminated in varying degrees by soil, organic matter, spores, viruses, bacteria and allergens, as well as aerosols such as smokes, dusts, fumes and mists. These contaminants may be introduced into the air from outdoors, or they may be returned to the air conditioning apparatus from within the space. The ease and efficiency with which they may be removed depends on the size, shape, specific gravity, concentration and surface characteristics of the particle.

Contaminant characteristics vary widely. Particle diameters range from molecular size up to 5000 microns*. Concentrations as high as 400 grains per 1000 cubic feet may be encountered. However, air conditioning applications usually involve the removal of particles no smaller than 0.1 micron in diameter and as large as 200 microns. Normal concentrations seldom exceed 4 grains per 1000 cubic feet. The specific characteristics of the particles to be removed are determined by the application. Thus, air purity control is a relative concept.

(*One inch equals 25,400 microns.)

The sizes of common contaminant particles are shown in Chart *10.* Typical outdoor air dust concentrations for various localities are noted in *Table 9.* Concentrations may increase during the heating season, especially in residential areas.

Particles of an oily nature with irregular surfaces electrostatically tend to agglomerate more readily. The settling and adherence of contaminants is, therefore, affected by other characteristics in addition to size and concentration.

TABLE 9—DUST CONCENTRATION RANGES

LOCALITY	DUST CONCENTRATION (grains per 1000 cu ft)
Rural and suburban districts	0.02 - 0.2
Metropolitan districts	0.04 - 0.4 (0.06 avg)
Industrial districts	0.10 - 2.0
Ordinary factories or workrooms	0.20 - 4.0
Excessively dusty factories or mines	4.0 - 400.

PERFORMANCE CRITERIA

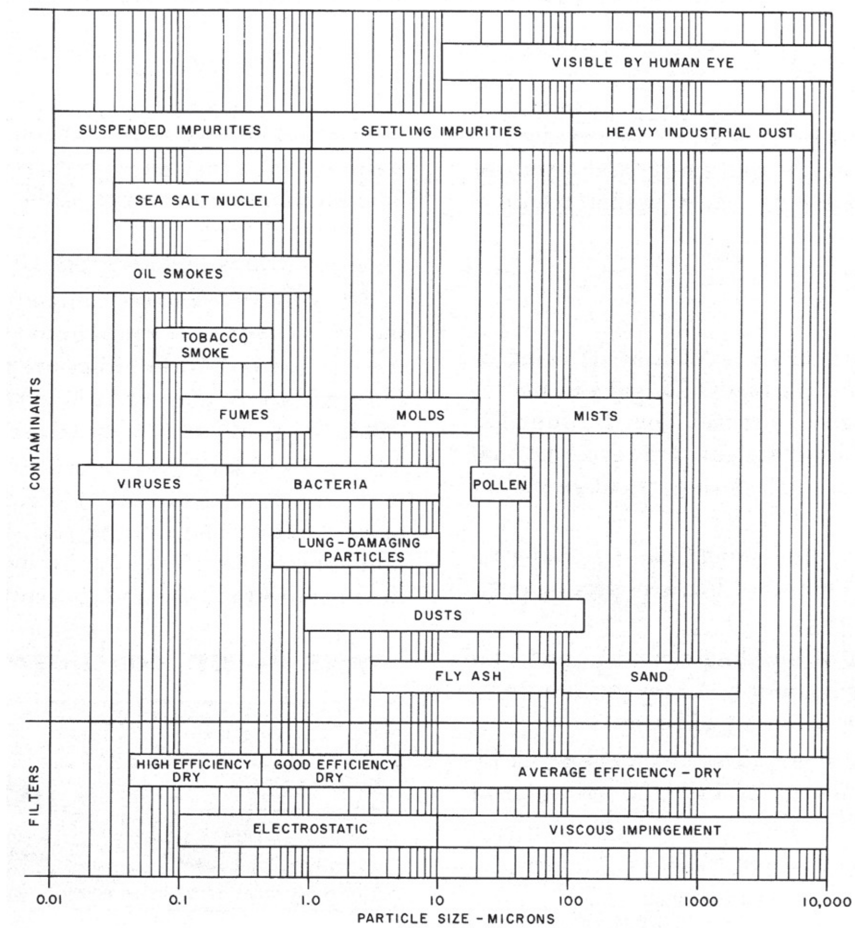
Atmospheric air cleaners (referred to as air filters) are rated in terms of efficiency (arrestance), resistance to air flow, and dust capacity. The three most critical performance factors are the following:

1. The variation of filter resistance with air flow.
2. The variation of filter resistance with dust load at design air flow.
3. The effect of dust loads at design air flow on filter efficiency.

Performance data for a typical unit filter are illustrated in *Fig. 54* Filter resistance increases with air flow (face velocity) or with dust load at. design air flow. The efficiency of a particular filter varies not only with dust load hut also with the characteristics of the contaminating particles. For this reason, *Fig. 5-4* specifies the test procedure used to rate the filter.

The capacity of a filter is a measure of its usable life prior to disposal, renewing or cleaning.

CHART 10—FILTER APPLICATION



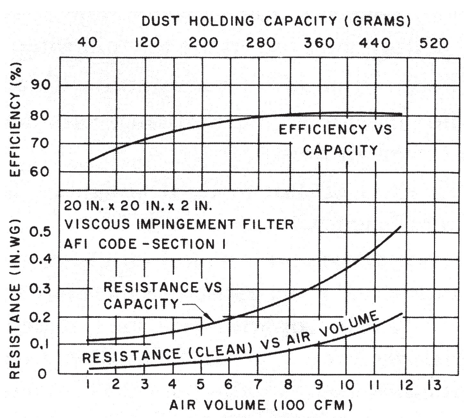


FIG. 54 — TYPICAL PERFORMANCE DATA (UNIT FILTER)

STANDARDS AND CODES

Air filter manufacture and installation should conform to the recommendation in Pamphlet. 90A of the National Board of Fire Underwriters and to all codes and laws applying at the job site.

The efficiency and capacity of air filters are determined by several standardized test methods, differing primarily in the test aerosol used and the method of measuring the amount of dust passed by the filter. The three most common test procedures are these:

1. The weight method, with test aerosols as specified by the Air Filter Institute Code and a modification of the former ASHVE Code.
2. The dust spot method, with procedures as standardized by the Air Filter Institute and the National Bureau of Standards.
3. The D.O.P.* test, a particle count method utilizing a chemical smoke aerosol.

* Di-Octyl-Phthalate

These three methods (lifer in application and the results are difficult to convert to common terms. In comparing the performance of various *filters* it is therefore imperative that the test used to obtain the published data l)e noted in each case.

The weight method expresses filter efficiency in terms of the particle weight removed, relative to the weight introduced to the air stream. It is particularly useful in evaluating the performance of mechanical filters of average efficiency. However, this test overstates filter effectiveness in removing small particles of light weight.

The dust spot test rates filters in terms of the relative opacity of stains on filter paper thin which the air is passed. The optical density of the spots are measured photometrically. This type of test is useful primarily in evaluating air cleaning devices of high efficiency, such as electronic air cleaners. In addition it provides a measure of filter efficiency in removing the sort of (lust most likely to cause discoloration of walls and ceilings. Test results are, however, sometimes inconsistent and are difficult to interpret.

The D.O.P. test relates filter performance to the light-scattering tendency of smoke particles approximately 0.3 microns in diameter. Measurements are made photo electrically. The test is used primarily to determine the ability of filters of very high efficiency to remove specific particles, such as pollen. It requires carefully controlled laboratory conditions and expensive equipment. It cannot be used to determine filter capacity.

TYPES OF AIR CLEANERS

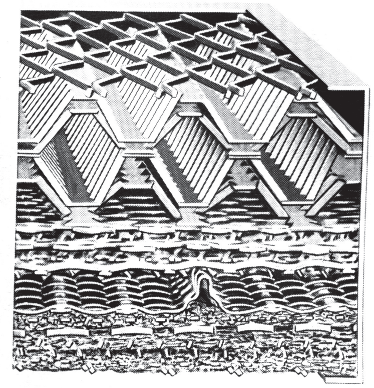
Viscous Impingement

Filters of the viscous impingement type utilize a filtering medium relatively coarse in texture and constructed of fiber, screen, wire mesh, metal stampings or plates. The medium is coated with a viscous substance such as oil or grease. As the many small air streams abruptly change direction thin the filter, contaminating particles are thrown against the medium where they adhere. Efficiencies of 65% to 80%, based on the weight method of testing, are achieved in the case of cleanable media.

This type of filter is available in a throwaway style or may be obtained with a replaceable medium, a manually cleanable medium, or an automatically renewed medium.

Media designed for filtering velocities of approxImately 300 feet per minute usually increase in density in the direction of the air flow. Thus, the larger particles are the first removed, prolonging filter life. This progressive density is illustrated in *Fig. 55.* High velocity filters operating at approximately 500 feet per minute are normally nondirectional and of uniform density. *Figure 56* shows a cleanable viscous impingement panel filter.

Automatic viscous impingement filters may be of the replaceable media or renewable media type. The former consists of a moving filter roll. The latter is constructed of overlapping filter panels attached to a moving chain and moving thru an oil bath. The self-cleaning filter is shown in *Fig. 57.* In either case the filter curtain may he actuated by a timing mechanism or a pressure sensing device. Automatic filters present a relatively constant resistance to air flow, while panel filter resistance varies considerably as the dust load increases. Automatic filter efficiencies vary from 80 % to 90 % , based on the weight method.

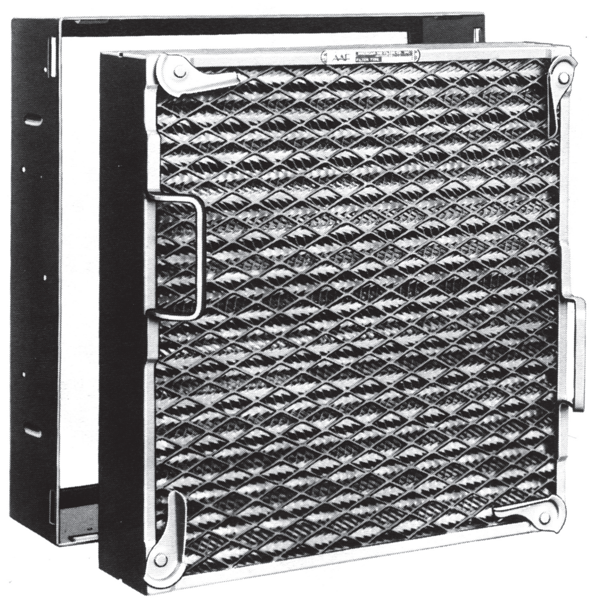


Courtesy of American Air Filter Co., Inc.

FIG. 55 — VISCOUS IMPINGEMENT FILTER SECTION

Dry Media

Dry filters consist usually of a permanent frame and a dry replaceable medium of cellulose, asbestos or glass fibers, specially treated paper, cotton batting , wool felt or synthetic material. The air passages thru the medium are



Courtesy of American Air Filter Co., Inc.

FIG. 56 — CLEANABLE VISCOUS IMPINGEMENT FILTER

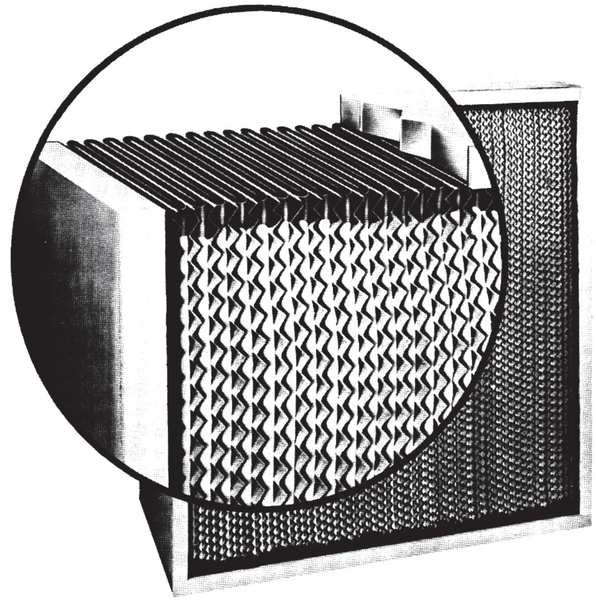
Courtesy of American Air Filter Co., Inc.

FIG. 57 – AUTOMATIC VISCOUS IMPINGEMENT FILTER

smaller than those of the viscous impingement type filter, and therefore lower air velocities are necessary to avoid excessive resistance . In order to obtain a large surface area relative to cross – sectional area, the medium is usually pleated in accordion form.

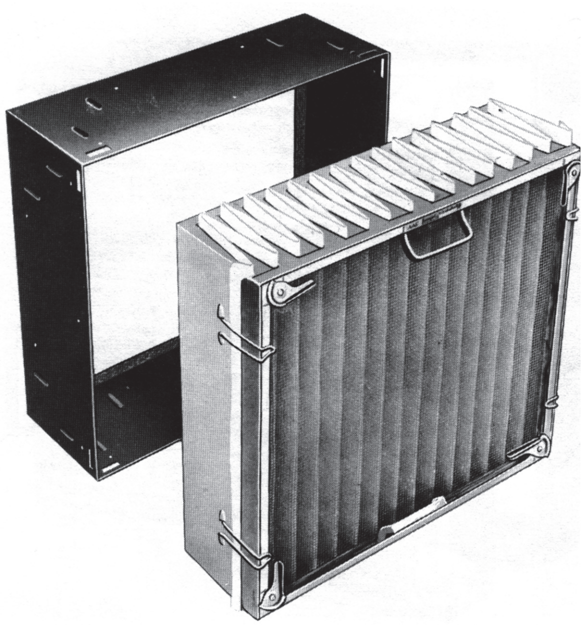
Figure 58 shows a medium efficiency dry filter with an area ratio of 7 : 1 Such a filter is capable of a wide efficiency range varying from 84 % to 95 % based on the AFI weight test, depending on the medium used.

Figures 59 and *60* illustrate vary high efficiency filters with area ratios of from 25:1 to 50:1. This type of filter may be obtained with and efficiency as high as 99.97 % , D.O.P. test method. D.O.P. efficiencies above 90 % are usual.



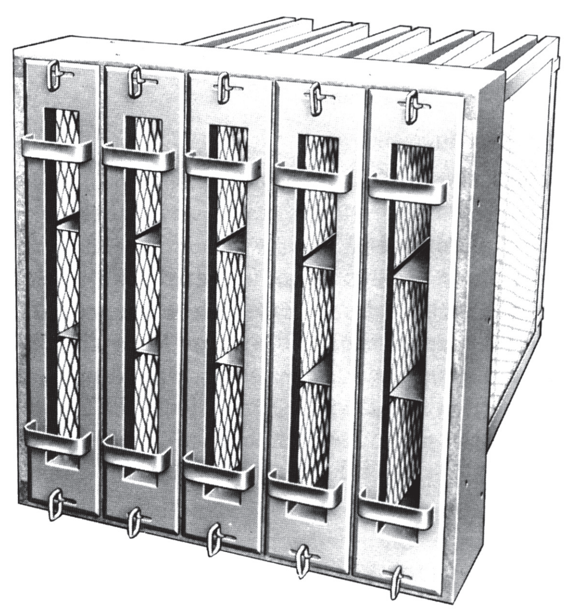
Courtesy of Cambridge Filter Corp.

FIG. 59 – HIGH EFFICIENCY DRY FILTER



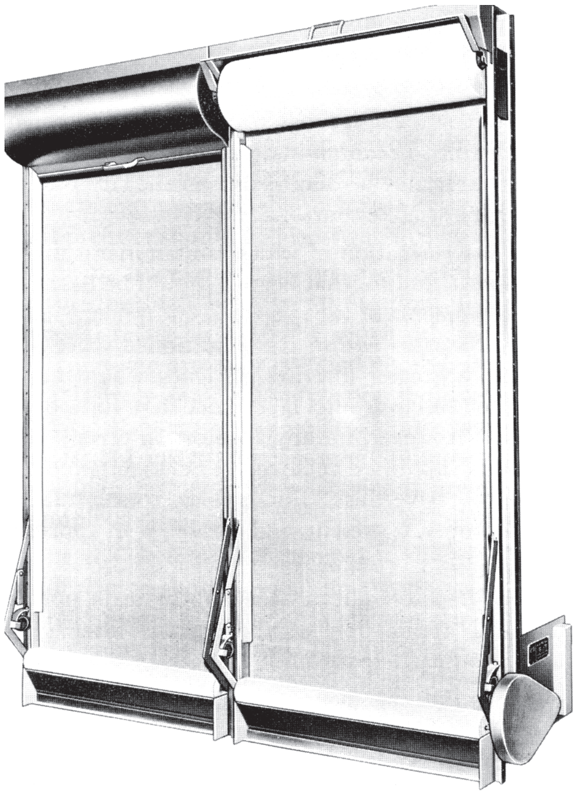
Courtesy of American Air Filter Co., Inc.

FIG. 58 – DRY FILTER CELL WITH FRAME



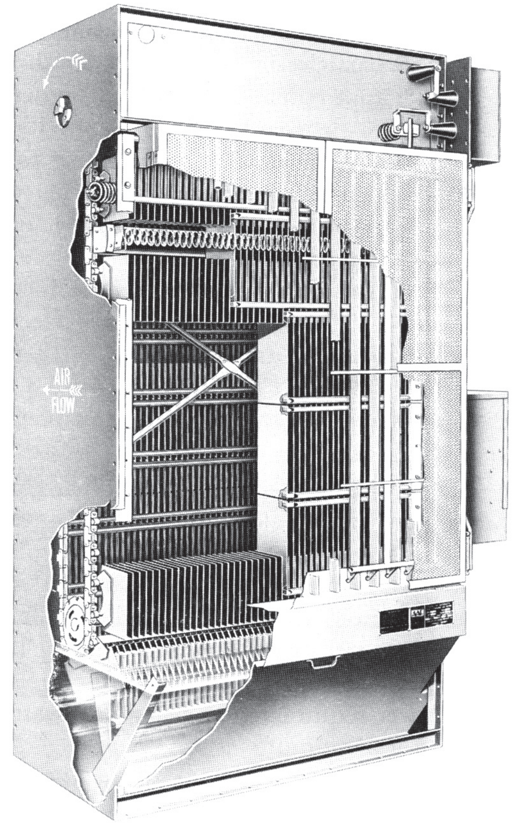
Courtesy of American Air Filter Co., Inc.

FIG. 60 – DRY FILTER CELL, POCKET TYPE



Courtesy of American Air Filter Co., Inc.

Fig. 61 — Automatic Dry Filter



Courtesy of American Air Filter Co., Inc.

Fig. 62 — Electronic Air Cleaner, Ionizing Type

Dry filters are available in automatic construction, normally utilizing a moving roll of disposable medium (*Fig. 61*). Movement may be controlled by a differential pressure sensing device. Thus , operating air resistance is maintained relatively constant.

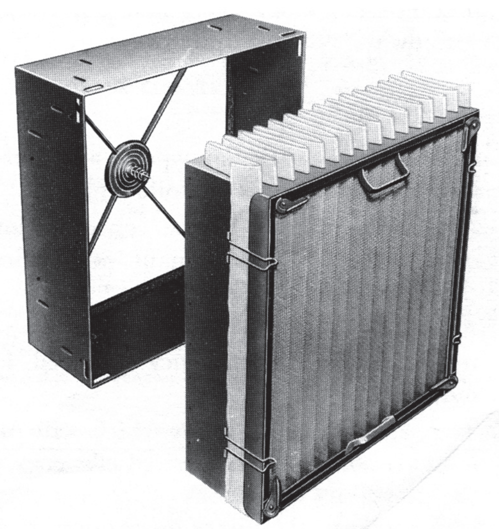
The efficiency of a dry filter depends on the size and spacing of the fibers in the medium used. Media with the smallest, most densely distributed fibers provide the highest efficiencies. High efficiencies, however, are usually associated with high resistance, short life and low dust holding capacity.

Electronic

Electronic air cleaners, often referred to as precipitators, are of two varieties: the ionizing type and the charged media type. They are illustrated respectively in *Fig. 62* and *63*.

The ionizing type of electronic air cleaner ionizes contaminating particles by passing the air thru an electric field of approximately 12,000 volts potential. The particles are then collected on charged plates which are usually coated with an adhesive to prevent re-entrainment of the particles. Efficiencies of 85% to 90%, based on the dust spot test, are achieved. The collecting stage operates at approximately 6000 volts. The high voltages are obtained from rectifiers supplied with 110/120 volt single phase electrical service. Power consumption varies from 12 to 15 watts per 1000 cubic feet per minute, with an additional 40 watts required o energize the rectifier tube heaters.

The charged media electronic air cleaner consists of a panel filter with an electrostatically charged dry medium.



Courtesy of American Air Filter Co., Inc.

Fig. 63 — Electronic Charged Media Filter

It therefore combines the principles of electronic precipitation and dry mechanical filtration. The produced efficiency averages about 60% by the dust spot test method. Approximately 12,000 to 1 3,000 volts are required to charge the dielectric medium. Power requirements are about 8 watts per 1000 cubic feet per minute.

Although the ionizing type of electronic cleaner may be obtained in replaceable cell construction, the precipitator shown in *Fig. 62* is automatically self-cleaning. Moving collector plates are cleaned and reoiled by the same method as employed with the automatic viscous impingement filter *(Fig. 57).* Semiautomatic cleaning is also available, utilizing nozzles for water cleaning and reoiling.

The resistance to air flow imposed by an ionizing type electronic cleaner is very small. For this reason the unit may feature screens or perforated plates at the air entering and/or leaving side to promote uniform air flow thru the precipitator.

APPLICATION

The choice of a particular type of air filter for a given application involves the following steps:

I. A determination of the size, concentration and characteristics of contaminants present in both the outdoor air and return air.
2. A decision regarding the size of particles to be remove and the efficiency required for removal.
3. The selection of the filter which will provide most economically the desired efficiency under the prevailing conditions of labor cost, power costs and annual hours of operation.

Air contamination may be appraised by costly laboratory analysis or by an estimation based on past experience and general data. The latter method is preferable in all except highly specialized applications.

Chart 10 and *Table* 9 may be used with judgment to determine air contamination. Additional data of local interest may be obtained from the city Bureau of Health or smoke control agency.

The determination of which contaminants are to be removed, and to what degree, should be based on the requirements of the processes, equipment, material or occupants within the conditioned space. For example, a greater filtering efficiency would be required for an electronics laboratory than for a bowling alley. However, for any application certain contaminants should be removed. These contaminants include abrasive (lusts, lint, pollen, concentrations of toxic fumes, ii present, and carbon, if in appreciable quantities.

Chart 10 also indicates the approximate normal ranges of application of the various filter types based upon particle size only.

Viscous impingement filters efficiently remove contaminating particles larger than 10 microns in diameter, particularly if the particles are oily. The coarse media use (1 are well suited to large particle sizes anti concentrations. The capacity and life of such filters are great and therefore maintenance is relatively inexpensive. High velocity unit filters (500 feet per minute) are not suitable to heavy lint applications since the media density is not progressive.

Dry media filters are more efficient than viscous impingement filters in removing particles in the submicron range. However, capacities are smaller with the finer media used. Filters of average to medium efficiency are useful in the collection of lint. High efficiency filters are intended primarily for removing $_1$)articles of small size and concentration. Dry filter life is relatively short, anti therefore, maintenance costs are usually higher than for impingement filters.

Automatic dry filters of the roll medium type are seldom used to remove atmospheric dust. They are well suited to the removal of lint as found in textile mills or dry cleaning establishments, and may be used for the removal of ink mist in the printing industry.

High efficiency dry filters *(Fig. 59 and 60)* are especially effective in the removal of viruses and bacteria, anti are therefore useful in hospital air conditioning. They may also be considered for protection from nuclear fallout and agents of chemical and biological warfare.

Electronic air cleaner efficiencies compete with the more efficient dry filters in the submicron particle size range. Only the self-cleaning ionizing type is suited to high contaminant concentrations since collector plates rapidly become less efficient as their dust load increases. Where high particle concentrations are encountered, the use of a viscous impingement prefilter should be investigated.

Ionizing type precipitators are useful on high pressure or high velocity applications for the removal of relatively fine dirt which otherwise tends to accumulate around discharge nozzles. Because of the relatively small maintenance requirements of this type of filter, it may be applied to large air tie-liveries and installations where equipment is relatively inaccessible or where service is infrequent or incomplete.

The characteristics of charged media air cleaners are similar to those of medium efficiency dry filters. charged media filters are less efficient than ionizing precipitators but electrical failure of the associated equipment does not completely destroy their usefulness. Operation at relative humidities exceeding 70% may adversely affect the dielectric properties of the medium. Individual resistors may be provided for each filter circuit to limit the

current flow thru the medium, should the medium become damp.

Regardless of the type of filter selected, the automatic self-cleaning feature renders servicing less dependent on the human element and provides a relatively uniform air resistance and air flow.

Outdoor air anti return air may he separately cleaned with different filter types if the characteristics of the contaminants to be removed are widely different.

Table 10 indicates the relative initial anti total annual costs of different types of air cleaning installations.

SELECTION

Filter size is usually determined by the rated air quantity per unit or panel as published by the manufacturer. These rated air deliveries are established with regard to practical air velocities as dictated by the characteristics of the medium employed. An overload of as much as 10% to 15% may be permissible, depending on the medium anti on the filter construction.

Table 11 is a tabulation of typical velocities and air resistances for various filter types. The air resistances are base(l on clean filters. Pressure drops reflecting a partially expended medium should be used for fan static pressure calculations, as recommended by filter manufacturers.

The sizes of filter units or panels are normally standardized anti limited in number. Installations are then built up from the basic units. Manually serviced viscous impingement filters are usually available in sizes 20 in. x 25 in., 20 in. x 20 in., 16 in. x 25 in. and 16 in. x 20 in. Standard thicknesses are 1 in., 2 in. and 4 in. Odd sizes are available only at considerably higher cost unless relatively large quantities are required.

Dry media filters are often available in only one size from each manufacturer, but some may offer a size selection. Charged media electronic filters are also of limited size selection.

Ionizing type electronic cleaners and self-cleaning filters are normally available in height increments of several inches but in standard widths limited to two or three sizes or combination thereof.

Some flexibility of selection may he exercised in the use of manually serviced mechanical filters by utilizing a "V' bank to obtain a greater ratio of filter area to cross-sectional area. This method is illustrated in *Fig. 64.*

Desirable characteristics for viscous impingement filter adhesive included homogeneity of film, a viscosity relatively constant with temperature change, a resistance to the development of mold spores and bacteria, a high ability to wet and retain dust at all temperatures, minimal evaporation, fire resistance, and freedom from odor.

TABLE 10—RELATIVE AIR CLEANING COSTS*

TYPE OF AIR CLEANER	RELATIVE COST PER 1000 CFM	
	Initial	Annual Owning and Operating**
Viscous Impingement		
Throwaway (2 in.)	0.55	1.45
Renewable (4 in.)	0.80	2.0
Cleanable (2 in.)†	1.0	1.0
(4 in.)	2.3	1.4
Automatic self-cleaning‡	3.6 - 8.0	1.9 - 3.0
Dry Media		
Cleanable and renewable (2 in.)	0.95	1.1
(8 in.)	3.3	1.5
High efficiency renewable	9.5 - 17.6	5.9 - 7.6
Electronic (ionizing)‡		
Plate or cell	12.4 - 21.3	2.9 - 5.9
Automatic	17.8 - 31.1	4.3 - 7.2
Electronic (charged media)‡	10.7 - 14.2	2.9 - 3.7

*3000 hrs per year.
†Basis of comparison.
‡Assumed minimum size of 10,000 cfm.
**Includes interest and depreciation.

INSTALLATION

Location

In an air conditioning system, air filters are usually located upstream of the fan, between the cooling coil and preheat coil, if any. This location simplifies duct and casing design for a built-up system, avoids the net static pressure loss associated with and acute transformation downstream of the fan, and produces a more uniform air distribution thru the filters. in addition, a measure of comfort is provided in the winter for the service attendant, and coils are protected from dust deposits and algae

TABLE 11—OPERATING DATA

TYPE OF AIR CLEANER	Nominal Velocity thru Media (fpm)	Resistance thru Clean Filter (in. wg)
Viscous Impingement		
Throwaway (2 in.)	300	0.06 - 0.12
Renewable (4 in.)	300	0.12 - 0.24
Cleanable (2 in.)	300 - 500	0.04 - 0.12
(4 in.)	300	0.08 - 0.20
Automatic self-cleaning	500	0.30 - 0.50
Dry Media		
Cleanable and renewable (2 in.)	60	0.08 - 0.13
(8 in.)	35	0.10 - 0.12
High efficiency renewable	5 - 20	0.50 - 1.20
Electronic (ionizing)*		
Plate or cell	300 - 400	0.15 - 0.30
Automatic	400 - 500	0.20 - 0.32
Electronic (charged media)	35	0.03 - 0.12

*Includes front and rear screens.

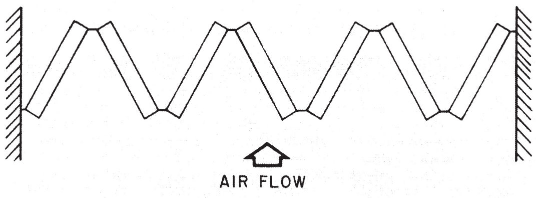


Fig. 64 — Unit Filter, "V" Bank

formation. Such a location minimizes the possibility of introducing rain or snow, a factor of great importance in the application of electronic air cleaners. In the case of viscous impingement filters, adhesive temperature variations are minimized.

If high efficiency air cleaners are used, the preferred location is downstream of the fan. Any air leakage thin the duct will be outward, and air cleanliness will thus be maintained. With any high efficiency filtering device, mechanical or electronic, located upstream of the fan, the duct or casing between the filter and the fan should be carefully caulked and the connections felted against leakage.

If no preheat coil is used, an electronic cleaning device should be located no closer to the outdoor air intake than the height dimension of the device.

Layout

The unsatisfactory performance of air cleaning devices can often be traced to improper installation practices or the lack of regular maintenance. Therefore, filter installations should be planned to meet engineering requirement and to facilitate service.

An inspection and service area of sufficient depth should be provided before and after the filter bank. A minimum access of two feet for viscous impingement filters and three feet for high efficiency dry filters is suggested. Electronic air cleaners may require five feet on the entering air side to permit the full opening of swinging doors or ionizer panels. The manufacturer's data should be consulted for more detailed information.

Access doors should be installed in the apparatus, upstream and downstream of the filter bank. In addition, ladders or catwalks are required for access to filter tiers at heights above six feet. Electric lights of the marine type facilitate service, and are suggested on both sides of the filter bank.

Viscous impingement and dry media unit filters are usually removed from the entering air side of a bank. However, they may be available for servicing from the leaving air side if such is specifically requested. The air how arrow should be observed in installing or replacing progressive density filters such as the viscous impingement type.

Duct and apparatus casing, both at the entering and leaving air sides of the filter bank, should be (designed and installed to insure even air distribution over the face of the filters. This is especially important in the case of electronic ionizing air cleaners or other low air resistance filters. For this reason perforated plates, grilles or screens are often installed upstream or downstream of electronic ionizing air cleaners. Manufacturers may included such baffling devices with the precipitator units.

Prefilters may be considered upstream of high efficiency dry media and electronic air cleaners if high dust or lint concentrations are present. These prefilters also serve to distribute the air uniformly.

Proper provision should be made for the collection and drainage of water if the filters are to be cleaned in place by hoses or nozzles.

Outside air intakes should be located at a height and position such that the introduction of heavy concentrations of surface or roof dirt, automobile fumes anti refuse is minimized. In take screens should be no coarser than 16 mesh. Louvers should be well constructed, particularly if the filter bank is near the intake.

A "V" or staggered filter bank may be employed to increase the ratio of filter surface area to cross-sectional area. In factory-built air conditioning units with filter boxes designed for low velocity filters, it may be necessary to blank off uniformly a portion of the cross-sectional area if high velocity filters are to be used. In this case, the use of a factory-built high velocity filter box is more appropriate.

Operation of electronic filters should be dependent upon electrical interlocks with apparatus access doors in the interests of operator safety. These interlocks interrupt operation as long as an access door remains open.

If sprinkler protection is required, piping must be provided from the building sprinkler system or the city water system. Many air cleaners feature built-in sprinkler provisions requiring only connection.

Mechanical air filter installations should include a draft gage or other differential pressure indicator to signal the need for cleaning or replacing filters or to warn of the failure of automatic filters.

MAINTENANCE

It is difficult to predict, on the basis of filter air resistance, when a manually serviced filter will require cleaning or replacement. Two indicators used to determine the need for servicing are a 10% decrease in air flow or an increase in resistance of two to three times the initial resistance. The intervals between cleanings vary with the application, type of filter and stage of job installation.

The rotation method of cleaning is often used, particularly on extensive installations. Under this method, only certain filter units are cleaned each week. Such a practice insures a more constant work load and a more uniform air resistance at any time.

The size of the installation dictates the most economic manual cleaning means. Filters may be cleaned in place with hoses or fixed nozzles on large jobs, while smaller installations may favor the use of a filter cleaning tank together with an appropriate number of spare filters.

Self-cleaning filters and precipitators should be observed for the expiration of disposable media or the accumulation of sludge in the collecting pan. Many manufactures provide signals for their equipment to indicate the need for service.

Manufacturers' recommendations regarding the method and interval of cleaning or replacing filters should be followed.

HEATING DEVICES

The heating devices commonly employed directly with air conditioning systems are designed to heat air under forced convection. They are usually located within the air conditioning apparatus and / or ductwork.

The media used for heating include steam, hot water, electricity and gas flame. In addition, for special applications, glycols and hot refrigerant gas may be used.

STANDARDS AND CODES

Methods of testing and rating forced circulation air heating coils utilizing steam or hot water are prescribed in ASHRAE Standard 33.

Various aspects of the construction and installation of electric heaters are dictated by Underwriters' Laboratories requirements. Installation of such equipment is also governed by the National Electric Code.

The installation and piping of gas-fired duct furnaces is prescribed by the American Standards Association Bulletin 21.30. Installation is also influenced by requirements of the National Board of Fire Underwriters. The manufacture of gas-fired apparatus is directed by standards of the American Gas Association.

The application and installation of all types of heating devices should also conform to local codes and regulations.

TYPES OF EQUIPMENT

Steam Coils

Steam heating coils consist of a series of tubes connected to common headers anti mounted within a metal casing. To insure efficient heat transfer, either plate type or spiral fins are bonded to the tubes mechanically or with solder. Tubing is usually constructed of copper, in standard tubing sizes up to and including one inch OD. Fins are often of aluminum with spacings ranging from three to fourteen to the inch. One-row and two-row coils are available with tubes spaced from one to three inches on centers. *Figure 65* illustrates a steam heating coil. The offset tubes provide for changes in length due to temperature variations.

Since the proper performance of steam coils depend on the uniform distribution and condensation of steam in the tube, several methods have been devised to insure this uniformity. Individual orifices may be built into the supply end of each tube, or distributing plates may be installed within the steam header.

Uniform steam distribution and leaving air temperature are also provide with the steam distributing tube type of coil. This design features a tube within a tube, with the inner tube perforated along its length. Steam is supplied to the inner tube and admitted thin the orifices to the outer or condensing tube. Condensate is collected in the return header. A steam distributing tube coil is shown in *Fig.* 66, and the principle is described in *Fig. 67.* In *Fig. 66* note that the tubes are pitched within the casing to promote the rapid return of condensate.

Steam heating coils are available in various tube lengths ranging from one foot to ten feet. Casing widths up to four feet may be obtained for a single coil.A

Hot Water Coils

Hot water heating coils are similar in construction, size and appearance to single tube steam coils. Although comfort heating systems seldom require hot water coils of more than two rows, greater depth of surface is available. Fins are usually spaced from a minimum of seven to a maximum of fourteen to the inch. A hot water beating coil is shown in *Fig. 68.*

In order to provide optimum combinations of capacity and water side pressure drop, various circuiting arrangements are employed. On multiple-circuit coils, tabulators are sometimes installed within the tubes to produce the turbulent flow necessary for efficient heat transfer.

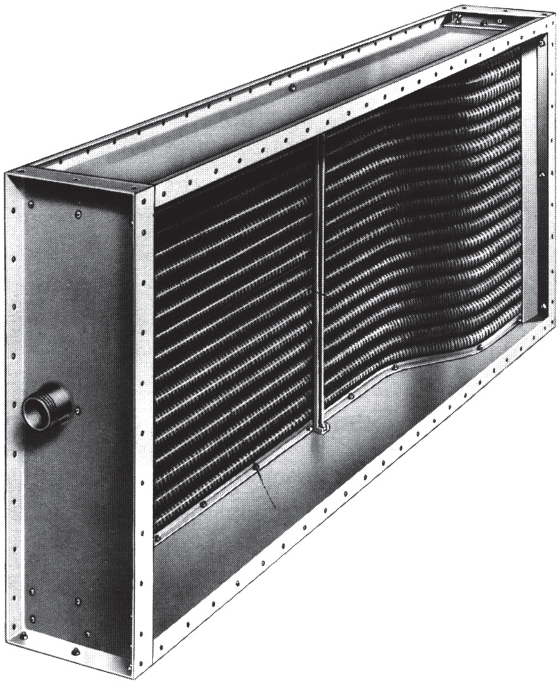
Electric Heaters

Electric heating devices are commonly available in either the open type or finned tubular type. These are illustrated in *Fig. 69 and 70.* The open type consists of a series of electrical resistance coils framed in a metal casing and exposed directly to the air stream. The finned tubular type of heater is made of finned steel sheaths

containing resistance wire surrounded by refractory material.

In order to achieve incremental control of heater output, multiple electrical circuits may be obtained. Although normal applications seldom require more than three circuits, as many as required are possible.

Standard voltages include 115 volt, single phase, and



Courtesy of Aerofin Corp.

FIG. 65 — STEAM HEATING COIL

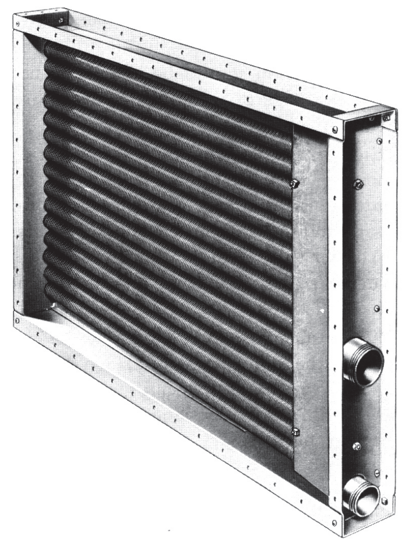
208 and 230 volts in single or three phase. Heaters are also available for operation on 440 and 550 volt service. Direct or alternating current control voltages may he specified.

Duct Furnaces

Gas-fired furnaces are built for installation in air ducts and in some packaged air conditioning units. *Figure 71* shows a duct furnace, and *Figure 51* illustrates the use of a gas-fired furnace in a packaged unit. Such equipment consists of a burner assembly, a heat exchanger, a plenum and controls. Natural, manufactured or liquified petroleum gas may be used.

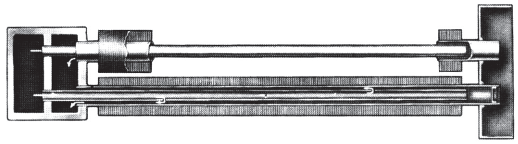
APPLICATION

Heating devices are used as preheaters and reheaters. A preheater is located upstream of the dehumidifier in an air conditioning apparatus, and is used either to raise the temperature of the entering air to a temperature above freezing or to supply the heat necessary for control of the temperature of the air leaving the dehumidifier. Both functions are often performed by a single heater. A reheater located downstream of the dehumidifier is used to control the temperature of the conditioned space when it is subjected to varying cooling loads. *Figure 36* indicates the approximate reheat requirement dictated by a particular design relative humidity and sensible heat ratio. A reheater may also be used as a booster heater, compensating for wide



Courtesy of Aerofin Corp.

FIG. 66 — STEAM DISTRIBUTING TUBE COIL

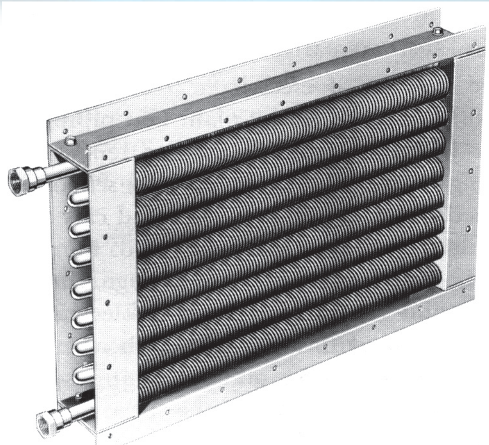


Courtesy of Aerofin Corp.

FIG. 67 — STEAM DISTRIBUTING TUBE PRINCIPLE

differences in cooling load characteristics between a particular zone and the rest of the space conditioned by an apparatus. If both functions are required, a central reheater mar be used to raise the supply air temperature to approximately room temperature or slightly higher. Booster heaters may then be installed in the branch ducts to the various spaces in order to provide control of room temperature.

Steam and hot water heating coils are most commonly employed for the applications cited. Steam coils are normally available for steam pressures up to 200 psig although special coils may be obtained for higher pressures. Hot water heating coils are used on low, mediun and high temperature hot water systems. However, applications involving water temperatures exceeding 300 F should be brought to the attention of the manufacturer.



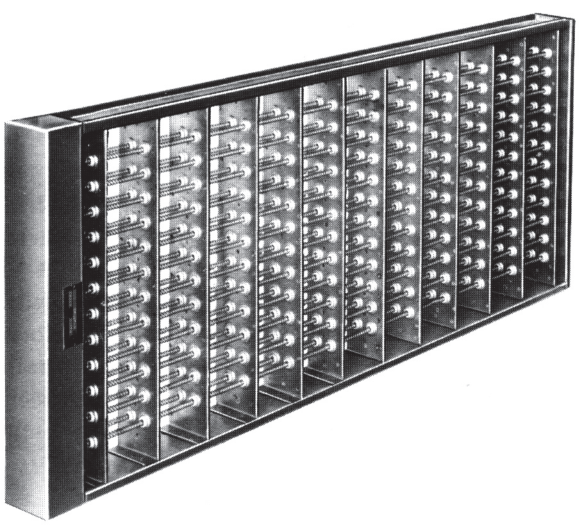
Courtesy of Aerofin Corp.

FIG. 68 — HOT WATER HEATING COIL

Steam coils of the distributing tube type are preferred over single tube steam coils an(l hot water (Oils for service where freezing air temperatures are encountered or where uniform heater leaving air temperatures are mandatory. Single tube steam coils and hot water heaters may, however, be used for preheat service if controlled as described under *Coil Freeze-up Protection.* A minimum entering water temperature of 150 F is suggested for hot water preheat service.

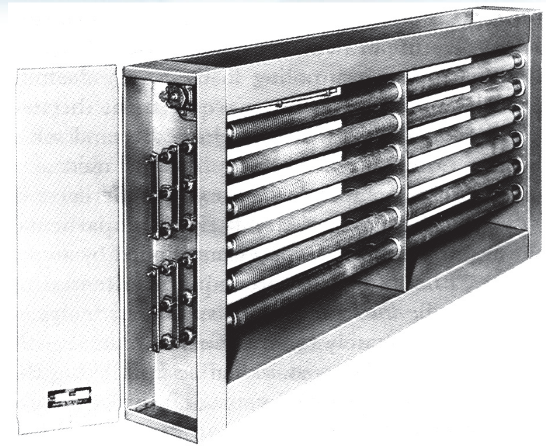
Plate fin steam and hot water coils are preferred to spiral for applications involving heavy concentrations of lint since they are more easily cleaned. If spiral fin coils are used for such applications, the widest appropriate tube spacing should be chosen.

Where corrosive substances are present in the air, steam or hot water, special coil materials are available. Most steam side corrosion problems may be avoided by the proper trapping and venting of noncondensables.



Courtesy of Industrial Engineering and Equipment Co.

FIG. 69 — OPEN ELECTRIC COIL HEATER



Courtesy of Industrial Engineering and Equipment Co.

FIG. 70 — FINNED TUBULAR ELECTRIC HEATER

The advantages of electric heating are low initial equipment and installation cost, a saving of floor space, compactness, simplicity of operation and control, a fast control response and cleanliness. Electrical facilities used in the summer for refrigeration equipment may be used in the winter for the heating system. At the same time electric heating has often proven costly to operate. For this reason it has been employed largely in mild climates or where electrical costs are particularly low.

Since the use of electric heaters eliminates the need for a central heating plant and piping system, electric heating can be used to tenant areas of shopping centers, department stores, schools, industrial facilities, banks, motels, railroad cars and markets. Electric heaters may be used for churches because of the short duration of usage, the low initial cost and the quick response. Electric heating may also be used in areas such as board rooms or executive offices where occupancy at night or on weekends may be common. It has been used successfully in conjunction with self-contained air

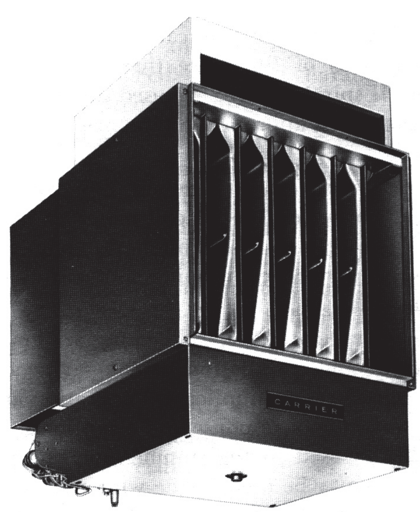


FIG. 71 — DUCT FURNACE

conditioning units anti as a source of auxiliary heat for heat pump systems.

The open type of electric heater operates at a temperature lower than that of the finned tubular heater, and therefore exhibits a longer life. It is lighter in weight, more rapid in response, offers less air resistance aid tends to cycle less. The finned tubular heater is particularity suited to applications where the heater may be subject to mechanical injury or where an explosion hazard exists. Stainless steel fins and sheaths are available for high humidities or corrosive a atmospheres.

Gas-fired duct furnaces may be used for preheat and reheat service. The advantages of such equipment are similar to those of electric heaters. Hence, duct furnaces may be used for similar applications in areas of relatively high power cost. As with electric heating the problem of coil freeze-up is not encountered. Gas-fired equipment should never be operated in corrosive atmospheres or in rooms where equipment should also be sufficiently removed from acid baths or degreasing tanks.

SELECTION

The selection of a heating device involves a consideration of the heating capacity required, the heating medium available or required anti its characteristics, the allowable resistance to the flow of air an(I/or heating fluid, the entering air temperature, the air quantity to be heated and the air velocity thru the device, dimensional limitations, installation requirements such as the type of control, special design requirements, and economy.

Steam and Hot Water Coils

The capacity of a steam or hot water coil of a given type may be increased not only by increasing the coil surface but also by increasing the coil face velocity thru reducing the face area. Since higher coil face velocities result in higher air side pressure drops, the selection of a coil surface may be more limited than at lower face velocities. Therefore, the size and capacity of a heating coil are interdependent, and each must be determined in relation to the other.

Minimum coil face area is usually determined by the design air quantity and a maximum allowable face velocity. The dimensions of the coil may then be chosen from among those dimensions available with the required face area. For a given face area, coils of greater tube length and smaller tube face are usually the least expensive. However, space requirements may limit both size and dimensions of a coil.

Coils are rated at face velocities of 300 to 1500 feet per minute. The maximum face velocity should be determined by the allowable air side pressure drop and the ambient sound level of the space served by the coil. Air pressure drops of 0.10 to 0.30 in. wg are suggested for preheat applications, while reheat coil friction may range from 0.15 to 0.35 in. wg.

Since heating coils do not condense moisture, and since no entrainment of moisture is possible, the face velocity of a heating coil mounted within a factory-built air conditioning unit should not be limited to the cooling coil face velocity.

The calculated heating load required of a coil is usually the primary determinant of the surface selected. Various combinations of fin spacing, tube spacing and coil depth result in a wide variety of available surfaces. The heat transfer capacity of a given surface varies directly with face velocity, steam pressure, entering water temperature or water tube velocity. ft varies inversely with entering air temperature.

Reheat coils are usually oversized. A 15% to 25% safety factor added to the calculated heating load provides for a rapid morning pick-up and compensates for duct heat losses. Steam preheat coils chosen to operate at subfreezing air temperatures with throttling control of steam should be undersized rather than oversized, if the required load cannot be met exactly. This practice reduces valve throttling at air temperatures of 25 F to 32 F, the range where excessive throttling most usually results in the freezing of condensate in the tubes.

When using duct reheat coils for large air quantities, it may be more economical to select a smaller coil to handle only a portion of the air, with the remainder being handled thru a fixed bypass around the coil. The air thin the coil is then heated to a higher temperature so that the mixture air is at the proper temperature. This may require a coil with more heating surface per square foot of face area, but in a smaller casing size. Assuming a coil face velocity, fin spacing and rows of coil, the coil air quantity is determined by dividing the required over-all temperature rise by the coil temperature rise, and multiplying by the total air quantity. The required coil face area can be found from the coil air quantity and the assumed coil velocity. The coil size is then selected to match closely the calculated face area. The coil bypass is sized as described in *Part 2,* and the dimensions are chosen to coincide with the coil casing length.

Figure 72 illustrates a steam coil selection table. Coil capacity may be expressed in terms of steam quantity condensed, heat transferred or final air temperatures alone. Hot water coil ratings may be similarly tabulated, except that, at each entering air temperature, capacities are listed for each surface at various entering water temperatures. Another method of presenting hot water heating coil ratings is illustrated in *Fig. 73.*

Heating coil performance ratings assume a rapid

elimination of air and other noncondensables and a uniform distribution of air thru the coil surface.

When steam coils are selected at face velocities exceeding those considered standard by the manufacturer, the amount of condensate per tube should be checked against the maximum recommended by the manufacturer. If the maximum allowable condensate per tube is exceeded, excessive steam pressure drops, water hammer and poor venting may result.

Electric Heaters

In addition to size and capacity an electric heater selection should specify electrical characteristics and the number of circuits required.

Electric heaters are usually chosen to fit a branch duct of given dimensions without requiring entering and leaving transformations. Therefore, face velocity is not the usual determinant of coil size, although for Underwriters' Laboratories approval a minimum face velocity must be maintained and uniform airflow provided. This minimum velocity is a function of entering air temperature and the total watts per square foot of duct area. Velocities may range up to 1800 feet per minute. Air side pressure drops are small compared to steam and water coil pressure drops, seldom exceeding 0.10 in. wg for an open type coil.

All of the electrical energy used in an electric heater is converted to heat. Thus, heater capacities in Btuh are determined by multiplying the kilowatt rating of the heater by 3412.

The number of circuits chosen depends on the degree and period of heating load fluctuations. The amount of control hunting permitted should be weighed against the economics of purchasing and installing the multiple circuit heater.

Duct Furnaces

Gas-fired duct furnaces are chosen according to the output satisfying the heating load. The required air temperature rise thru the furnace determines the air quantity to be handled by the device and the resulting air friction. If a greater branch duct air quantity is required, a fixed bypass may be provided as described under *Steam and Hot Water Coils.*

Atmospheric Corrections

Heating coil ratings are based on the standard atmospheric conditions of 29.92 in. Hg barometric pressure and 70 F. For significantly different air conditions such as at altitudes exceeding 2000 feet or at average air temperatures above 125 F, a correction should be applied to the required air temperature rise and the air quantity upon which the selection is based.

In determining the coil air temperature rise required or the heating load imposed by the admittance of air at temperatures below the design temperatures, such as thru ventilation or infiltration, the factor 1.08 should be adjusted in proportion to the ratio of air densities as found from *Chart 2.*

The design air quantity should be multiplied by the density ratio in order to determine the equivalent air flow at sea level. The adjusted air quantity and heating load (or air temperature rise) are then used to select a coil surface.

The coil size and face velocity are determined by the design air flow with no correction applied. However, the coil air side pressure drop should be corrected as described in *Part 2.*

Since the capacity of an electric heater does not depend on air quantity, no correction to the ratings is required. However, the heating load and air friction should be corrected as necessary.

Gas-fired duct furnaces employed at elevations exceeding 2000 feet should be derated in output by 4% for each 1000 feet above sea level.

COIL FREEZE-UP PROTECTION

The exposure of hot water or steam preheat and reheat coils to subfreezing temperatures, either by accident or intent, makes possible the freezing of water accumulated within the tubes and thus costly damage. The prevention of such occurrences requires consideration of the problem in the apparatus design and layout, in the selection of equipment, and in the choice of control methods.

The primary requirement for positive freeze protection is the assurance of uniform coil leaving air temperatures.

5 LB STEAM — 227 F

ENT AIR TEMP (F)	COIL SURFACE	COIL FACE VELOCITY (FPM) 300		400		500		600		700		800	
		Final Temp (F)	Cond*	Final Temp (F)	Cond*	Final Temp (F)	Cond*	Final Temp (F)	Cond*	Final Temp (F)	Cond*	Final Temp (F)	Cond*
0	A	43.5	*14.6*	39.1	*17.5*	36.0	*20.2*	33.8	*22.8*	31.9	*25.0*	30.2	*27.3*
	B	62.5	*21.0*	56.7	*25.5*	52.4	*29.4*	49.7	*33.5*	47.3	*37.2*	45.0	*40.5*
	C	76.0	*25.6*	68.0	*30.6*	62.0	*35.0*	57.5	*38.9*	54.0	*42.5*	51.0	*46.0*
	D	108.9	*36.8*	100.6	*45.4*	93.2	*52.6*	88.1	*59.7*	84.4	*66.6*	80.6	*72.7*
	E	125.0	*42.3*	114.6	*51.8*	107.1	*60.4*	101.2	*68.5*	96.3	*76.0*	92.4	*83.5*
40	A	75.8	*12.0*	72.2	*14.4*	69.7	*16.6*	67.9	*18.8*	66.3	*20.6*	64.9	*22.1*
	B	91.5	*17.4*	86.7	*21.0*	83.2	*24.2*	81.0	*27.6*	79.0	*30.5*	77.1	*33.3*
	C	102.6	*21.0*	96.0	*25.2*	91.1	*28.8*	87.4	*32.0*	84.5	*35.0*	82.0	*38.0*
	D	129.7	*30.2*	122.9	*37.4*	116.8	*43.4*	111.6	*48.4*	109.6	*55.0*	106.5	*60.0*
	E	143.0	*34.8*	134.5	*42.6*	128.3	*49.9*	123.4	*56.4*	119.3	*62.8*	116.1	*68.6*

*Condensate in pounds per hour per square foot face area.

FIG. 72 — TYPICAL STEAM COIL RATINGS

Air temperature stratification may be caused by incomplete mixing of outdoor and return air or by an uneven temperature rise thru the coil.

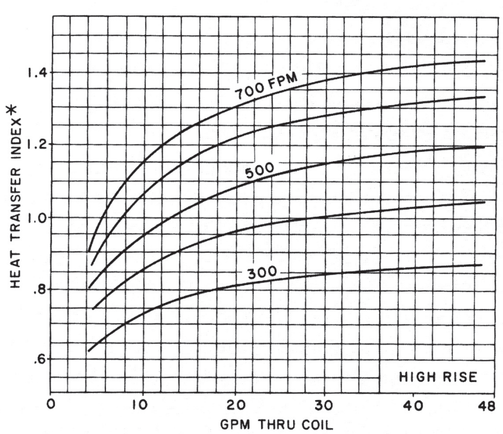
Where the mixing of outdoor and return air takes place upstream of a heating coil, mixing should be promoted by introducing the denser cold air at the top of the plenum and by providing as much airway length as possible. If a steam coil is employed, the steam should be supplied from the naturally colder side of the plenum.

If the mixing of outdoor and return air is to occur downstream of a preheat coil, it is suggested that only the minimum outdoor air be heated and that the maximum outdoor air dampers be maintained closed at subfreezing temperatures. If efficient downstream mixing has been provided, the preheat coil may be used instead to heat the return air to a temperature predetermined to yield the desired mixture air temperature.

Where a steam preheat coil served by a modulating valve tempers outdoor air, freezing of condensate in the tubes occurs most often at entering air temperatures in the range of 25 F to 32 F. Within this range the coil is usually operating under a severe partial load. The relatively small amount of steam admitted to the coil condenses completely before the end of the tube is reached, resulting in stratification. For this reason, if modulating control of steam is required at subfreezing entering air temperatures, the use of the steam distributing tube type of coil is suggested.

Single tube steam and hot water coils may be used for the tempering of subfreezing air, but the heating medium should not be modulated at entering air temperatures below 35 F. However, in cold climates such a design may produce overheating. To provide a degree of control while avoiding stratification, two preheat coils in series, each furnishing a part of the required capacity and controlled in sequence, may be employed. An alternative method consists of the use of face and bypass dampers controlled by a plenum thermostat. The bulb of such an instrument should be located well downstream of the heating coil if space permits. Otherwise, an averaging type bulb should be used.

These same methods of obtaining control without stratification should be considered where steam distributing type coils of large capacity are used for preheat service. However, rather than employing two coils in series, it may prove economically preferable to utilize one coil with two control valves piped in parallel. The first valve to open may be sized to pass the minimum



$$*HTI = \frac{\text{heating load (1000 Btu/hr)}}{t_{ew} - t_{edb}}$$

FIG. 73 — TYPICAL HOT WATER COIL RATING CURVES

steam quantity necessary for even distribution of steam thru the tubes at a signal from a two-position outdoor air thermostat.

As mentioned previously, a minimum entering water temperature of 150 F is suggested for tempering subfreezing air with hot water. Uniform leaving air temperatures should be insured as described above. In addition, a safety control closing the outdoor air damper at entering water temperatures below 150 F is suggested.

Another method of freeze protection is the circulation of an inhibited glycol solution thru a water coil. A two-row coil with a single circuit is the best protection against stratification. The system should be designed to supply the glycol solution to the coil at a temperature of about 150 F at peak conditions with a high temperature drop of about 50 degrees. The steam valve to the glycol heat exchanger is controlled by the air temperature leaving the coil.

Another requirement for adequate freeze-up protection of steam coils is the positive and complete drainage of condensate from the tubes. Any type of steam coil may be damaged if condensate is allowed to accumulate and freeze thru poor design of the system or coil. For this reason an ideal position for a steam preheat coil is with the tubes vertical anti with the condensate header at the bottom. For either horizontal or vertical air flow, steam preheat coils installed with tubes horizontal should be pitched downward toward the condensate header to facilitate drainage. Many steam distributing tube coils feature tubes internally pitched for either horizontal or vertical air flow. In this case installation is simplified, and the only precaution necessary is to make sure that the condensate header is lower than the steam header if air flow is vertical.

Positive condensate drainage is also insured by the proper design of the condensate return system. Adequately sized steam traps and vacuum breakers are among the most important design considerations. Refer to the discussion in this chapter under *Layout* and to *Part 3*

Larger heater tube diameters provide more positive condensate drainage and more uniform leaving air temperatures.

The outdoor air dampers of an apparatus should be closed whenever the fan is not running. In this way the induction of cold air by stack effect thru an inoperative coil is minimized.

Steam preheat coil control valves, if used, should be of the “normally open” type. Such valves should be sized to provide the maximum required capacity at a large pressure drop. Since valve capacity varies as the square root of the pressure drop, the valve tends to be undersized if steam pressure falls, and freezing is less likely to occur within the coil.

Although occurring less frequently, the freezing of reheat coils may be a problem, particularly if preheaters are not employed. If such is the case, the same provisions as described for preheat coils should be considered if complete mixing of outdoor and return air cannot first be guaranteed. Where face and bypass control of a dehumidifier is employed, the preheat coil should be located so that the air bypassed to the reheat coil is tempered as well as the dehumidified air.

INSTALLATION

Location

In an air conditioning apparatus the preheater is usually located between the outdoor air intake and the filters. Reheaters are mounted downstream of the dehumidifier coil, either within the apparatus or in the ducts. The latter location is often chosen where more than one control zone is served by a single air conditioning unit.

Duct-mounted heating devices may be located outdoors as well as indoors. Heater and duct should be externally insulated and weatherproofed. As much of the steam condensate return system as possible should lie within the heated space. Terminal boxes for electric heaters should be weatherproof.

Layout

Hot water coils and single-pass steam coils of both the single tube and steam distributing types may be installed with tubes horizontal or vertical and used for vertical or horizontal air flow. Multi-pass steam coils designed for use also with hot water are limited to horizontal tube applications. Regardless of the orientation, steam coils should be mounted so that the condensate connection is below the steam connection.

Steam and water coils may be assembled in banks. Coils so mounted should be supported individually in angle iron frames, thus protecting the lower coils from damage and facilitating coil removal.

Sufficient access space should be provided around a heater to permit maintenance and removal. Connections to ductwork should be so constructed to allow coil removal without disturbing the duct. Duct access doors on either side of the coil permit cleaning of the equipment in place. Refer to *Part 2* for a description of the design of ductwork surrounding a heater.

A fixed heater bypass may be located around or below the heating surface. A single-acting bypass damper with blades inclined toward the leaving air side of the heater promotes the mixing of heated and bypassed air.

The design of hot water and steam coil piping is

described in *Part 3.* Hot water coils should be piped so that the water enters at the bottom connection, and coil vents should be provided as required. In a steam coil installation where the condensate return main is higher than the coil steam trap, a condensate pump, lift trap or boiler return trap should be used to move the condensate to the main. A minimum of 18 inches should be maintained between a steam coil condensate outlet and the floor to provide space for traps and piping.

In the design of heater installations considerations should be given to the prevention of air temperature stratification. Uneven air temperature rises thru a heating coil may result not only in coil freeze-up problems but also in the supplying of air of non-uniform temperature to branch ducts splitting off downstream of the heating surface. Stratification may be minimized by the proper design of duct splits, by the use of two coils mounted in parallel and supplied from opposite sides of the apparatus and, if necessary, by the use of individual duct heaters. A horizontal supply air split is suggested with single fan air handling units, while a vertical split is more appropriate for multi-fan units. Other measures to reduce stratification such as the use of steam distributing tube coils are outlined in the section dealing with coil freeze-up protection.

The location and layout of electric heaters and gas-fired duct furnaces relative to surrounding combustible surfaces is limited by applicable standards and codes.

When locating gas fired heaters, avoid locations which have a positive exhaust unless provision is made to make up air to provide for the exhaust as well as the combustion air.

When locating electric heaters in equipment in front of fan motors, overheating of the motors may result because of the high temperatures that can be obtained.

Duct furnaces may be grouped in series or parallel. Outdoor air of approximately 14 cubic feet of air per cubic foot of gas should be provided. Flue design should be in accordance with American Gas Association standards.

CONTROL

The capacity of a heating coil may be varied in accordance with the load by control of the flow of the heating medium, by air volume control, or by air bypass control. The control of steam or hot water flow is most commonly employed. A multi-zone air conditioning apparatus may utilize air bypass control. If steam is used in this case, "on-off" control of the coil is preferred to minimize stratification.

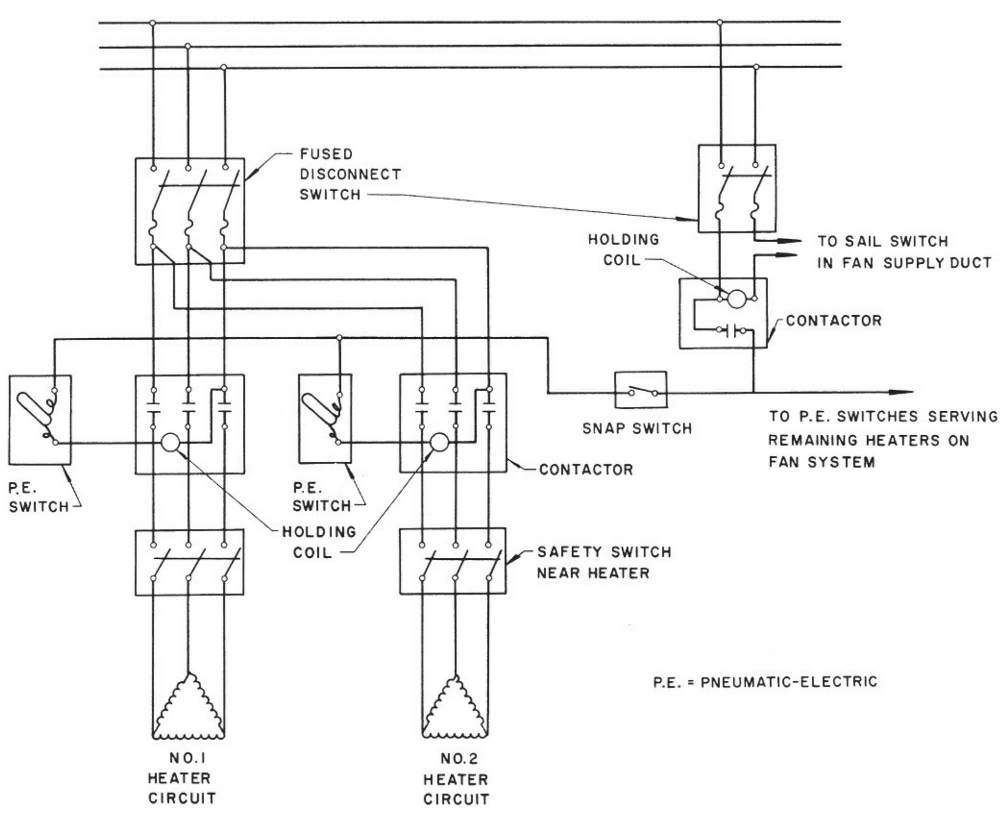


Fig. 74 — Electric Heater Control

Where a reheat coil has been selected with excess capacity as suggested above, the use of two control valves mounted in parallel and furnishing respectively one-third and two-thirds of the steam required may improve the accuracy of control at relatively low heating loads.

Figure 74 illustrates the wiring and control of electric heaters. The controlling instrument shown is a pneumatic-electric switch actuated by a pneumatic zone thermostat. Electric thermostats may also be utilized. The supply duct sail switch insures that the heaters operate only when the fan is running. On a single-heater fan system, a thermal switch may also be used for this purpose. Where multiple circuit heaters are employed, individual pneumatic-electric switches and contactors are used for each circuit. A step thermostat may be used in place of the pressure switches.

Gas-fired duct furnaces require safety controls such as gas valve low voltage control, a normally closed gas valve, a high bonnet temperature cutout, a pilot safety control, and a gas pressure regulator for other than LP gas.

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